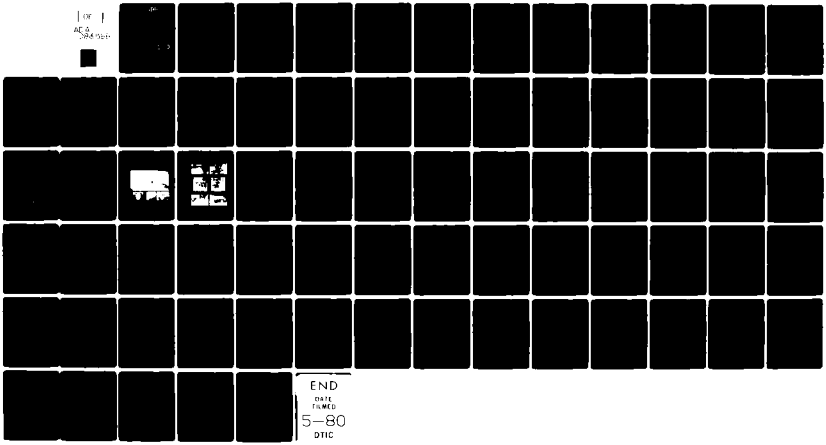
D-A083 558 DAVID W TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CE--ETC F/G 13/10
THE DEVELOPMENT AND EVALUATION OF A RUDDER ROLL STABILIZATION S--ETC(U)
MAR 80 A E BAITIS MIPR-Z-70099-4-52249-A
UNCLASSIFIED DTNSRDC/SPD-0930-02 NL



DAVID W. TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER

Bethesda, Md. 20084

THE DEVELOPMENT AND EVALUATION OF A RUDDER ROLL STABILIZATION SYSTEM FOR THE WHEC HAMILTON CLASS

by

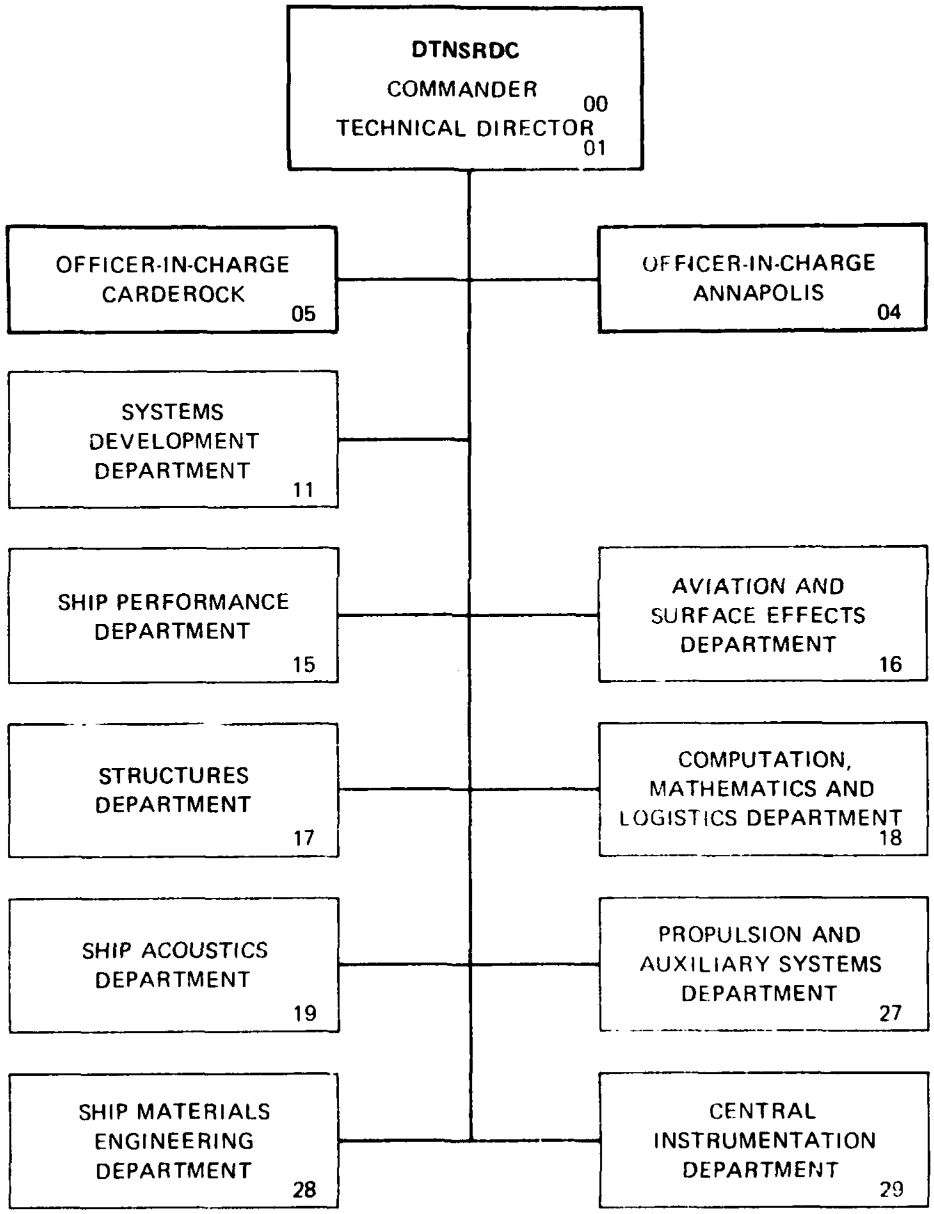
A. E. Baitis

APPROVED FOR PUBLIC RELEASE: DISTRIBUTION UNLIMITED

SHIP PERFORMANCE DEPARTMENT

March 1980 DTNSRDC/SPD-0930-02

MAJOR DTNSRDC ORGANIZATIONAL COMPONENTS



UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER DTNSRDC/SPD-0930-02	2. GOVT ACCESSION NO. AD-A083 558	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) THE DEVELOPMENT AND EVALUATION OF A RUDDER ROLL STABILIZATION SYSTEM FOR THE WHEC HAMILTON CLASS		5. TYPE OF REPORT & PERIOD COVERED Final 1975-1980
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) A. E. Baitis		8. CONTRACT OR GRANT NUMBER(s) MIPR-Z-70099-4-52249-A
9. PERFORMING ORGANIZATION NAME AND ADDRESS Ship Performance Department David W. Taylor Naval Ship R&D Center Bethesda, Maryland 20084		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS (See reverse side)
11. CONTROLLING OFFICE NAME AND ADDRESS United States Coast Guard, G-DST-2/54 2100 Second Street, S.W. Washington, D.C. 20590		12. REPORT DATE March 1980
		13. NUMBER OF PAGES 75
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office)		15. SECURITY CLASS. (of this report) UNCLASSIFIED
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

APPROVED FOR PUBLIC RELEASE: DISTRIBUTION UNLIMITED

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

Technical assistance in the preparation of this report was provided by The Scientex Corporation under contract with the David W. Taylor Naval Ship Research and Development Center. The contract number was N00167-80-M-1274.

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Rudder Roll Stabilization	Steering Motor
Ship Steering System	Loads
Roll Stabilization System Development	Steering System Wear
Simulation of Ship and Steering System	Rudder Rates
Full Scale Trials	Rate Saturation

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

A major thrust of recent research and development effort in surface ship dynamics has been in the area of ship/aircraft interfacing. This is due to the increased importance of helicopters and vertical/short takeoff and landing aircraft as the major combatant capability of nonaviation naval ships. One ongoing effort in the ship/air interface area is the joint U.S. Navy/Coast Guard Rudder Roll Stabilization Program. This program has been directed at using a ship's existing rudder(s) to control roll in a seaway.

(Continued on reverse side)

DD FORM 1 JAN 73 1473 EDITION OF 1 NOV 65 IS OBSOLETE

S/N 0102-LF-014-6601

UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

324674

(Block 10)

USCG MIPR Z-70099-4-52249-A
USN Task Area ZF 61412001, Element 62766N, Work Units 1568-123 and 1568-124
USN Task Area SF 43411212, Element 62543N, Work Units 1504-100 and 1504-200
USN Element 63504N, Project W0570SL, Work Unit 1568-819

(Block 20 continued)

This report summarizes the steps taken to develop a rudder roll stabilization system for Coast Guard cutters of the HAMILTON Class, including a chronology of the problems encountered and the steps taken to overcome these problems. The performance features of this system for a wide range of operating conditions are documented, and critical design features and system limitations and benefits are identified.

Preliminary results from the operational evaluation of two preproduction prototype systems indicate that the system, as designed, is very effective for short duty cycle utilization, such as in typical helicopter and vertical/short takeoff and landing aircraft operations. The system is also very effective at reducing roll over long periods of time, so as to increase habitability. However, the reliability of existing steering systems needs to be determined before the roll reduction system can be utilized for long duty cycles.

TABLE OF CONTENTS

Page

LIST OF FIGURES

LIST OF TABLES

NOTATION

b_j	Cosine-squared spreading function
K_2	RRS rate gain
RRS	Rudder roll stabilization
$S_\zeta(\omega)$	Spectral density function
T_o	Modal wave period
W_{EXC}	Roll excitation simulation moment
W_r	Rudder roll moment
W_w	Wave induced roll moment
WBK	With bilge keel
WOBK	Without bilge keel
α	Swash plate angle
δ	Rudder angle
δ_A	Actual rudder angle
δ_C	Rudder command
δ_E	Error signal
δ_H	Helm command
δ_S	Roll control signal
$(\tilde{\zeta}_w)_{1/3}$	Significant wave height
λ_o	Wavelength corresponding to T_o
σ	Standard deviation
ϕ	Roll angle
$\dot{\phi}$	Roll rate
χ_j	Wave direction
ω	Wave frequency

ABSTRACT

A major thrust of recent research and development effort in surface ship dynamics has been in the area of ship/aircraft interfacing. This is due to the increased importance of helicopters and vertical/short takeoff and landing aircraft as the major combatant capability of nonaviation naval ships. One ongoing effort in the ship/air interface area is the joint U.S. Navy/Coast Guard Rudder Roll Stabilization Program. This program has been directed at using a ship's existing rudder(s) to control roll in a seaway.

This report summarizes the steps taken to develop a rudder roll stabilization system for Coast Guard cutters of the HAMILTON Class, including a chronology of the problems encountered and the steps taken to overcome these problems. The performance features of this system for a wide range of operating conditions are documented, and critical design features and system limitations and benefits are identified.

Preliminary results from the operational evaluation of two preproduction prototype systems indicate that the system, as designed, is very effective for short duty cycle utilization, such as in typical helicopter and vertical/short takeoff and landing aircraft operations. The system is also very effective at reducing roll over long periods of time, so as to increase habitability. However, the reliability of existing steering systems needs to be determined before the roll reduction system can be utilized for long duty cycles.

ADMINISTRATIVE INFORMATION

This work was performed by the staff of the Ship Performance Department, Code 1568, of the David W. Taylor Naval Ship Research and Development Center. Funding was provided by the U.S. Coast Guard under MIPR Z-70099-4-52249-A, the U.S. Navy IR/IED program under Task Area ZF 61412001, Element 62766N identified as Work Units 1568-123 and 1568-124; the U.S. Navy Seakeeping Research and Development Program under Task Area SF 43411212, Element 62543N identified as Work Units 1504-100 and 1504-200; and the Navy Vertical Takeoff and Landing program under Element 63504N, Project W0570SL identified as Work Unit 1568-819.

INTRODUCTION

A major thrust of the research and development in surface ship dynamics over the last decade has been in the area of ship/aircraft interfacing. The importance of this area of naval ship design and use has grown substantially as the combatant capabilities of helicopters and vertical/short takeoff and landing aircraft have become more and more apparent. These aircraft now represent the major combatant capability of nonaviation naval ships. Furthermore, as the use of large conventional carriers and associated fixed wing aircraft become increasingly restricted for political and fiscal reasons, the importance of nonaviation ships with aircraft capability continues to grow. As a consequence, the David W. Taylor Naval Ship Research and Development Center (DTNSRDC) has been actively engaged in developing, in cooperation with other Navy and Coast Guard activities, operational ship/aircraft system design procedures and hardware to bridge the ship/aircraft interface.

This ship/aircraft interface is strongly dependent on weather and ship motions. Therefore, roll reduction is desirable to minimize the possibility of aircraft dynamic rollover during landing or lateral sliding on the deck when the craft is not secured prior to takeoff or subsequent to landing. Rudders have long been recognized as potential antiroll devices. However, no practical Rudder Roll Stabilization (RRS) system had ever been developed because of the possibility of interference between roll reduction and the primary rudder role as a steering mechanism.[1,2,3,4*]

In 1974, DTNSRDC initiated a program to examine the possibility of installing RRS systems on U.S. Navy ships. The hydrodynamic characteristics and maximum rudder rates of a wide range of naval vessels were used as baseline data. From the analysis performed in this study, DTNSRDC concluded in 1975 that RRS was feasible for many but not all categories of vessels, at least from the point of view of available maximum rudder roll moment.[*]

---

*A complete list of references is provided on pages 67-69.

**Vessels examined included CV-59, DLG-26, DD-963, DD-692, DE-1052, DE-1040, FFG-7, DE-1006 and PG-84.

Concurrent with DTNSRDC's initial study, the U.S. Coast Guard cutter HAMILTON was undergoing ship/helicopter interface trials in an effort to develop wind envelopes appropriate for the operation of the Coast Guard's large H-3 helicopter from the relatively narrow flight deck of HAMILTON.[5] These trials showed that roll stabilization was essential to increase the safety of H-3 landings and takeoff operations from HAMILTON Class cutters. Roll stabilization with antiroll fins or tanks was determined to be uneconomical. Therefore, using the steering system for roll stabilization was an attractive, economical alternative. A joint USN/USCG RRS program was subsequently initiated in the Fall of 1975.

SYSTEM DESCRIPTION

The premise of the RRS program was that roll stabilization is a secondary function of the rudder. Its primary purpose is to steer. However, as shown in Figure 1, the rudder produces simultaneously a roll moment (roll arm x lift) along with the yaw moment (yaw arm x rudder lift) needed to change ship course.

Typical ship response periods to a roll moment are about 8 to 12 seconds, whereas typical response periods to a yaw moment are about 30 to 35 seconds. The significant differences between these response periods permit the simultaneous superposition of yaw and roll control signals on the rudder without adversely affecting the response in either mode.

RRS involves opposing the wave-induced roll moment by the rudder roll moment which varies according to ship speed and rudder angle δ. Stabilization is attained by adding a properly phased roll control signal δ_S to the steering or helm command δ_H. This is achieved by making δ_S proportional to roll rate $\dot{\phi}$ so that it anticipates roll motion ϕ by 90 degrees.

STEERING SYSTEM

The steering system installed aboard the USCGC HAMILTON Class is shown schematically in Figure 2. This system utilizes twin rudders which are mechanically coupled through rudder tiller arm actuators (steering rams). These steering rams, which are connected to the rudder stocks and determine rudder angle, are driven by hydraulic flow from the steering

pumps. The pumps, in turn, are powered by electric motors. Steering pump output is controlled by the angular position of the pump swash plate. The hydraulic flow illustrated in Figure 2 is directed such that the trailing edge of the rudders will move to port.

For reliability, two independent hydraulic steering pumps and motors, two swash plate control actuators, two sets of steering cables and two sets of control electronics are included in the system design. The redundancy continues up to the Jered differential, which mechanically drives the swash plates of the port and starboard steering pumps. These port and starboard systems can be operated individually or together. When operated together, the single pump rudder rate of 2.33 degrees per second is effectively increased to 4.66 degrees per second.

RRS DESIGN GOALS

The goal of the RRS development program was to establish the feasibility of RRS for U.S. Navy and Coast Guard applications. This goal was to be attained by demonstrating the effectiveness of an RRS system on a HAMILTON Class cutter. A hardware oriented approach to the roll stabilization requirement of HAMILTON was selected to expose both the advantages and the disadvantages of the RRS system. This approach was also expected to lead to the development of RRS as an alternative, economical roll reduction system that could be retrofitted to existing naval combatants or installed in new ships under construction, where the ship mission is such that the larger, more expensive roll reduction capability of anti-roll fins is not required.

In order to reduce the roll motion of a ship by its rudder, the steering system must meet certain economic, hydrodynamic and machinery criteria, which also apply to other lift dependent stabilization systems such as antiroll fins.

The economic criteria concern minimizing the cost, maintenance, and allocated space within the ship relative to the stabilization system. These criteria can be met by using as much of the existing expensive steering system machinery as possible, avoiding most of the capital and maintenance costs associated with conventional roll stabilizers.

Two hydrodynamic criteria apply. First, the rudder generated roll moment should be 14 percent or more of the wave induced roll moment W_w in a Sea State 4 if roll is to be substantially reduced.* Second, RRS should not create over 2 degrees of vessel yaw or otherwise adversely affect the helmsman's steering task.

Three machinery criteria apply. Two are related to the rudder geometry (size, location, shape) and the rate of rudder movement. Specifically, machinery lags, as well as the maximum rudder rates, must be of suitable magnitude to permit efficient use of the rudder moment for roll stabilization. The third machinery criterion is that the steering system reliability must not be decreased and the system wear rates must not be unduly increased.

The various criteria were translated into six RRS design goals:

1. Short duty cycle stabilization (as needed for aircraft takeoff/ landing)
2. Independence of the RRS system and the autopilot
3. Simplicity and reliability
4. Low initial cost
5. Low life cycle cost
6. Long maintenance interval

By restricting RRS system operation to short duty cycles (less than one-half hour), the wear requirement of a candidate steering system is reduced. By specifying RRS autopilot independence, the design and construction of an operating RRS system is simplified. The additional goals specified make the RRS system economically attractive.

In the design of the RRS system, the maximum available rudder rate of the existing steering gear was not altered from design specification, even though 10 to 15 percent additional rudder rate could have been obtained with the installed machinery. However, the duration of the availability of this maximum rate was increased substantially. This design decision was made to minimize steering system lags while retaining the same maximum loads on the critical rudder stock and tiller arm

---

*See Table C-1 for definition of Sea States.

bearings as when the rudder is used only for steering. However, the frequency of occurrence of these maximum loads was increased.

The effect of this increased maximum load occurrence is that the steering system wear is accelerated. The impact of the accelerated wear is mitigated by limiting the use of the RRS system to short duty cycles of less than one-half hour.

RUDDER ROLL STABILIZATION

The RRS system developed by DTNSRDC adds three basic control components to the existing steering system of the ship: (1) bridge control unit; (2) electronic controller; and (3) swash plate actuator. These are illustrated schematically in Figure 3. Figure 4 presents a photograph of these components and illustrates their location aboard ship. Additional photographs are provided in Figure 5, where the steering system and RRS components are shown installed aboard ship. RRS systems are relatively inexpensive because the system components modify control mechanisms rather than the steering machinery itself.

The bridge control unit functions as the RRS system on/off switch. The DTNSRDC electronic controller measures and superimposes the helm command δ_H with the roll stabilizer command δ_S to arrive at the rudder command δ_C. This command is compared with the actual rudder angle δ_A to obtain the error signal δ_E. This signal activates the pump swash plate control actuators, which determine the rate and direction of flow into the steering system rams, ultimately leading to a change in rudder angle.

The DTNSRDC swash plate control actuators are coupled to the steering system swash plate actuators through a mechanical fail-safe clutch on a common shaft, permitting either the RRS control signal (as described above) or the direct helm command to determine steering pump flow. This clutch is designed to engage the direct helm command whenever the RRS system is turned off or fails.

PROTOTYPE DEVELOPMENT

USCGC HAMILTON, selected for prototype development, is 378 feet (115 meters) in length, displaces 3,000 tons (3,049 tonnes) and meets the economic, hydrodynamic and machinery criteria concerning suitability for

rudder roll reduction. Ship/helicopter interface trials aboard HAMILTON in April 1975 established the desirability of roll stabilization. For example, it was observed that crew movement on the flight deck was restricted due to excessive roll and associated lateral accelerations while underway, even in a mild, frequently occurring Sea State 4. These crew movement impairments would have increased the risk of operational problems had aircraft operations been conducted. Time histories of ship motions where this risk was increased are shown in Figure 6. It was necessary for the crew to use handrails on the flight deck during the most severe motions recorded in this figure.

Since it was desirable to validate the results of the recently completed (1975) feasibility study, a joint USN/USCG effort was initiated to develop an RRS system for the October 1975 HAMILTON sea trials. The development effort began with the determination of the rudder moment capacity of HAMILTON Class cutters. The effort continued with the design, installation and performance evaluation of prototype hardware and software systems necessary for RRS. Finally, in 1979, operational evaluations were conducted aboard two HAMILTON Class vessels. A detailed description of the prototype development is provided in Appendix A.

RRS SYSTEM PERFORMANCE

SYSTEM SIMULATION PRIOR TO DEPLOYMENT

Prior to the January 1978 deployment of the preproduction prototype units aboard MELLON, extensive system performance tests were conducted using the roll-sway table at the DTNSRDC Antiroll Tank Facility.[6] These tests were conducted to establish the reliability of the RRS components and to document, for the at-sea operators, the expected roll reduction performance under all types of sea conditions, headings and speeds. A description of the test results under various sea state, heading, and speed conditions is given in Appendix B, and a discussion of the sea simulation technique is provided in Appendix C.

The most significant result obtained from the simulations was the determination that roll reduction increases proportionally to increased rudder moment capacity. Thus, increasing rudder moment generation capacity may significantly enhance RRS performance, provided that control

system limitations are not exceeded. Figure 7 illustrates this simulation result in quartering seas where roll is significant.

For retrofitted RRS systems on existing ships, rudder moment capacity can be increased by increasing the rudder rate, and thus its excursion. This excursion increase can be obtained by simply changing the steering motors and pumps. For ships in the design process, the rudder moment capacity can be increased by changing the rudder size, location and/or geometry.

SYSTEM PERFORMANCE DURING SEA TRIALS

Summaries of the June 1976 MELLON and the September 1979 JARVIS sea trial results are provided in Tables 1 and 2, respectively. (Bilge keels had been fitted to both cutters prior to these trials.) The tables present the ship responses and steering system responses separately. Yaw is considered to be a steering system response and represents fluctuation from the measured course. Ship responses are given in terms of significant single amplitude responses (the average of the one-third highest amplitudes). Steering system responses are presented in terms of size of the fluctuations from the mean of the signal (the standard deviation). Because the statistical properties of the steering system responses are not known, the extreme values of the responses cannot be inferred from the standard deviation.

June 1976 Trials Aboard MELLON

The June 1976 trials were conducted primarily in Sea State 4. Measured results are somewhat higher than those predicted by the simulation results (see Figure B2, Sea States 4 and 5). Roll reduction is about 33 to 40 percent for the ship with bilge keels in beam seas at 15 knots. As the ship heading shifts toward quartering seas, the measured roll reduction decreases to about 22 to 27 percent. This trend of decreasing roll reduction as heading shifts to quartering seas agrees with the trends of the simulation results in Figure B2 only when modal wave periods are 9 seconds or greater.

Pitch, as expected, is essentially unaffected by the RRS system. Similarly, yaw is not noticeably affected by the system although the

helmsmen reported that it was easier to steer with the RRS system activated.

September 1979 Trials Aboard JARVIS

The September 1979 JARVIS trial results further document the roll reduction performance of the RRS units currently undergoing operator evaluation. Ship speed ranged from 13.5 to 15.2 knots. Unlike the earlier MELLON trials in mild seas, these trials were conducted at essentially all headings in seas ranging from Sea State 4 to Sea State 6. The observed seas usually contained at least one train of long swell coming from a different direction than the wind seas. The headings in Tables 1 and 2 represent the headings into the largest observed sea or swell.

Because most trial conditions were in Sea State 5, the effect of sea severity on RRS performance was only observed in quartering seas. At these headings roll reduction decreased from about 28 percent in a Sea State 4 to 8 percent in a Sea State 6. This agrees with the simulation results presented in Appendix B.

Generally, roll reduction was maximum in near-beam seas with reductions ranging from 31 to 49 percent. The higher reductions occurred in the lower seas (as predicted by simulation results). Roll reduction was lower (22 percent) in head/bow seas than in comparable quartering seas (28 percent).

The lateral accelerations were measured with an accelerometer mounted rigidly to the flight deck at the bullseye. These accelerations were reduced when the RRS system was activated but to a lesser degree than roll itself. The skid-inducing lateral accelerations were frequently larger than the vertical accelerations. Neither pitch nor the associated vertical acceleration of the bullseye were consistently affected by the RRS system. Also, yaw generally was not affected by the RRS action, although rudder activity was clearly increased when the RRS system was operational.

Effect of Control Law Variations

The results of the simulations conducted in 1977 indicated that certain variations in the control law which governs the operation of the

RRS system could be beneficial to system performance. This observation was substantiated further by the analytical work of Whyte.[7] Therefore, during the JARVIS trial, experiments were conducted with control laws other than just roll rate to determine their impact on the RRS system effectiveness.

When the control law contained roll acceleration as well as roll rate, steering difficulty increased and, particularly in quartering seas, unacceptably large steering instabilities were introduced. After verifying this conclusion by a series of experiments with different helmsmen, the possible use of roll acceleration as a component of the control law was abandoned.

Attention was focused next on evaluating the potential benefits of adding a roll angle component to the stabilizing signal. Several experiments were performed with a roll angle component added to the control signal. The results are quite interesting. When the control law contains roll angle as well as roll rate, the gains in steering ease due to RRS operation (with roll rate only) are diminished. The roll reduction benefits of adding a roll component to the control signal in beam seas are negligible (compare run numbers 33 and 35), whereas the benefits are quite substantial in quartering seas (compare run numbers 37 and 38) where the RRS roll reduction improves from about 8 percent to 21 percent (see Table 2). In bow seas, however, the situation is reversed (compare runs 39 and 41) so that the roll angle addition reduces roll reduction from 26 percent to 2 percent. These results are confirmed by the motion time histories presented in Figures 8 through 10. In Figure 8, the roll and yaw motions are greater in bow seas when both roll angle and roll rate are used. Figure 9 indicates that there is little difference between the effectiveness of the two control laws for roll reduction in beam seas, though yaw was less when both roll angle and rate were used. Figure 10 reveals that stabilization with a roll angle and roll rate control law is more effective at reducing roll in quartering seas, though the yaw motion is increased, indicating an increase in steering difficulty. It is concluded therefore, that, unless the controller can alter the control signal mixture to account for the changes in roll introduced by heading changes, it is preferable to use simple roll rate control law. The controller aboard JARVIS was adjusted accordingly.

It is evident from these results that although controller efficiency can improve the roll reduction performance of the RRS system, a single, constant gain ratio between control signal components cannot be effectively employed. Only an adjusting multistage controller is likely to improve the RRS system performance (when the rudder rate or moment generating capacity cannot be altered). Manually, this two-stage controller would use a control law incorporating both roll angle and roll rate in quartering seas, and only roll rate in bow or beam seas, as determined by the ship operator. Automatically, this could be achieved by using a controller capable of employing the appropriate control law (with or without roll angle) as determined from the measured unstabilized roll.

FUTURE OF RUDDER ROLL STABILIZATION

There are two distinctly different kinds of RRS systems. The first provides short duty cycle stabilization for the performance of certain mission-critical tasks. The second provides long duty cycle roll stabilization analogous to that provided by antiroll fins[8,9] or tanks.[6,10] The first requires modifications to the control mechanisms of an existing steering system, while the second entails modifications to the steering system itself. In short duty cycles, the extra wear experienced by the existing system appears to be acceptable from the points of view of wear rate and maintenance. It has not yet been determined, however, what effect the use of a short duty cycle RRS system has in long duty cycle applications.

SHORT DUTY CYCLE RRS SYSTEMS

Short duty cycle RRS systems should be considered primarily for retrofitting to existing ships, especially those which do not have antiroll fins or tanks and whose successful mission performance requires short term roll stabilization. These systems are less attractive in new construction during the design stage, because long duty cycle systems can be incorporated easily and economically.

One area for improving the short duty cycle RRS of the HAMILTON Class consists of developing a more sophisticated electronic controller. Current results show that measurable improvements in roll reduction can

be obtained when the control law is adapted optimally for the seas and headings encountered. With the advent of digital controllers, such as the type specified for the FFG-7 fins,[11] optimal adaptation of the stabilization control law will become possible in the near future. This increased controller efficiency will, of course, also benefit long duty cycle RRS systems.

There is a possibility that short duty cycle RRS systems can be used for periods of time longer than one-half hour, provided the increased wear rates and machinery maintenance needs are acceptable. This was the case aboard MELLON when the system was used continuously for 56 days before the starboard electric motor burned out. It has not been determined whether this failure was due to the RRS system or related to overall usage/failure patterns aboard MELLON.

RRS IMPACT ON STEERING SYSTEM LIFE

In order to quantify and understand the effect of RRS operation on the life of the steering system, the steering pump motors were examined in some detail. The steering pumps themselves, however, whose (swash plate component assembly) wear rate is likely to be substantially increased with RRS usage, were not examined due to the absence of failures of these units during MELLON's 56 days of constant RRS usage.

The steering motors were examined in two stages: the rated power and heat rise capacity were determined and the true power and casing temperature were measured aboard JARVIS. An examination of the nameplate information and operating details of the motor were obtained from visits to HAMILTON and from DTNSRDC electric motor reference files. The steering motors were manufactured in 1965 as three phase, 440 volt 286 UN frame units with class H insulation. These motors are rated at 20 horsepower (14.9 kw) maximum continuous duty units while drawing 26 amperes. The motors have an intermittent, varying or 30 minute duty cycle rating of 30 horsepower (22.4 kw) while drawing 39 amperes. The heat rise capacity of the motors is 50°C above ambient. The important distinction between these two types of motor ratings must be kept in mind when considering the measured power and temperature of the motors as well as their conventional use.

Measurements were made when the RRS system was engaged (roll stabilized, both steering motors operating) and disengaged (roll unstabilized, only the starboard motor operating). Figure 11 shows the variation of the mean power of the starboard motor for changes in rudder activity. In the unstabilized mode, the mean power in all but one case exceeds the continuous duty power rating by 10 to 15 percent although it represents only about three-fourths of the short duty cycle power rating. Thus, the starboard motor in this conventional use actually exceeds the continuous motor power rating. When the RRS system is activated, the mean power increases to about 10 percent above the short duty cycle power rating, which is substantially above the continuous motor rating. (The total power used for stabilization is actually twice this amount, because the port motor is also operating). Therefore, rudder roll stabilization is attained by drawing on the order of 8 kw additional power out of each steering motor. These units were shown to be rugged and reliable because they withstood 56 days of continuous usage aboard MELLON. These observations may also indicate the degree of conservatism inherent in the rating of these motors.

The effect of the RRS-induced increase in mean power on motor casing temperature is shown in Figure 12. This figure only includes typical data for trials conducted in one day in nontropical waters once the motor had attained a stable operating case temperature. Ambient air temperature in the aft steering compartment was about 85°F, and each measurement was made over a one-half hour period. In each case, the heat rise capacity limit of the motors (about 175°F) was exceeded. This again illustrates the rugged quality or conservative rating of these motors. Furthermore, these results reveal that at least up to the short duty cycle power rating, the casing temperature does not vary significantly with mean power, and hence with RRS operation. Casing temperatures during operations in tropical waters exceeded the (200°F) range of the temperature sensor. Therefore, no specific data is available that indicates the effect of RRS operation on casing temperature in those warmer waters.

In an effort to extend the life of existing steering motors on HAMILTON Class cutters, DTNSRDC in the second half of FY 79 undertook a program to increase substantially the motors' air cooling systems to

prevent their armatures from burning out when the RRS system is deployed in tropical waters. This was done in two stages. First, aboard JARVIS, three fans, together with a shroud, were installed around each motor to increase cooling air flow around and through the motor casing. This did not increase motor cooling significantly. Therefore, aboard MELLON, the paint was stripped off of the central portion of the motors, and the surfaces were refinished with thermally conducting flat black paint (see Figure 5). In addition, the cooling fan capacity was increased. These steps led to an increase in cooling air exhaust temperature of 35°F, indicating a significant increase in motor cooling. A more detailed discussion of motor cooling is given in Appendix D.

LONG DUTY CYCLE RRS SYSTEMS

Long duty cycle RRS systems should be considered in the initial ship design process. The incremental cost of properly sized rudders and steering machinery would be very small, and it is unlikely that the improved rudder roll moment capacity would degrade the yaw moment capacity.

Before long duty cycle RRS systems can be employed routinely by marine engineers as alternative stabilization systems, a practical upper limit of usable rudder roll moment capacity must be established. The author, therefore, recommends that the steering system of an existing HAMILTON Class cutter be modified such that rudder rates on the order of 10 degrees per second for single pump operations can be achieved. The use of HAMILTON Class cutters is recommended because they are close in size to most naval combatants which could potentially benefit from RRS systems. In addition, cutters deployed off the West Coast routinely spend extensive periods of time in areas where severe seas occur frequently.

DTNSRDC further plans to test the potential of short duty cycle RRS systems for longer term use by developing appropriate systems and retrofitting them to a smaller USCGC class (the 1000-ton, 210-foot, medium endurance cutter of the RELIANCE Class). This development program is currently underway with system installation scheduled for Summer 1980. This program will provide a clearer indication of the overall effect of

the RRS system on the steering system. The problems aboard MELLON may have been atypical since other operational results indicate that the magnitude of the loads imposed by use of the RRS system did not affect rudder stock bearings, tiller arm bearings or other mechanical components of the steering system.

Guidance in the use of the RRS systems installed aboard the USCG cutters JARVIS (WHEC 725) and MELLON (WHEC 717) as well as the technical documentation required to repair and/or duplicate these systems is provided in Reference 24.

CONCLUSIONS AND RECOMMENDATIONS

From the results of the RRS development program, the following conclusions have been reached:

1. The primary objective of the RRS program has been achieved. That is, the feasibility of the RRS systems has been demonstrated. Roll reductions up to 50 percent have been measured.
2. The extra wear experienced by steering systems during RRS operation in short duty cycles appears to be acceptable, though the effect of extra wear in long duty cycles has not been established.
3. The effectiveness of RRS systems is directly dependent on available rudder roll moment. The available moment, in turn, depends on rudder rate. RRS system performance is rudder rate limited.
4. The available rudder rate dependent roll moment on HAMILTON is marginal and establishes a lower limit on practical rudder moment for RRS. In marginal systems, a significant improvement in RRS can be achieved by increasing the available rudder rate and/or by using optimal control logic.
5. Preliminary results indicate that RRS operation does not appear to affect steering motor temperature and thus motor life. Methods for increased motor cooling appear to be effective.

The results and conclusions of the RRS program lead to the following recommendations:

1. Short duty cycle RRS systems should be considered for retrofit applications.
2. Long duty cycle RRS systems should be considered in initial ship design, once a practical upper limit of usable rudder roll moment capacity has been established.
3. A digital controller should be integrated into the RRS system to optimize the adaptation of the stabilization control law.
4. When the performance of retrofit RRS systems aboard USCG cutters has been evaluated for reliability, endurance, wear rates and maintenance requirements, the U.S. Navy should seriously consider retrofitting as a viable alternative to the current apparent policy of not stabilizing ships at all (except for the FFG-7 class).
5. To eliminate problems involving motor overheating, HAMILTON Class vessels utilizing RRS systems should, if possible, replace the existing 30 hp steering motors with 40 hp units.
6. In lieu of the installation of new motors, the existing motors should be modified to provide for increased cooling. Before the modification, additional steering motor temperature measurements with and without increased cooling should be made during RRS system operation in tropical waters to verify that the methods used for increased cooling are adequate.
7. Once the effectiveness of the proposed cooling methods is established, the additional cooling hardware should be installed on all vessels utilizing RRS systems where larger motors cannot be used.
8. Maintenance records for all aspects of the steering system should be collected by U.S. Coast Guard 14th District Engineering staff for both JARVIS and MELLON to establish the impact of RRS operation on long term reliability.

ACKNOWLEDGMENTS

During the five years of RRS development, many people made significant contributions to the evolution of the prototype system. The author wishes to acknowledge the efforts of Messrs. W. McCreight and G. Cox of

DTNSRDC in the establishment of the feasibility of the RRS system. The support provided by Mr. T. Milton of USCG in developing a joint USN/USCG RRS program is acknowledged with appreciation, as is the assistance provided by Mr. G. Vogel of Jered Industries with regard to the establishment of HAMILTON steering gear characteristics and their adaptation for RRS.

The author also wishes to thank Messrs. J. Ware and R. Nigon of DTNSRDC for their support in the application of adaptive control theory and the development of the electronic control hardware.

The encouragement provided by Captain H. Olsen of MELLON in 1976, and the enthusiastic support of the crews of HAMILTON, MELLON and JARVIS is acknowledged with thanks, as is the assistance of Lt. Commanders D. Freezer and M. Chappell of USCG R&D Headquarters. In particular, we wish to acknowledge Lt. Commander Chappell's steady and continuing encouragement without which the successful completion of this work would have been doubtful.

The author is grateful to Mr. G. Minard of DTNSRDC for his effort in the construction, installation and evaluation of the RRS system, to Mr. W. Meyers for his effort in the development of the simulation, and to Dr. A. Powell of DTNSRDC for his penetrating reviews. Also, the efforts of Mr. T. McNamara are noted with regard to the preparation of Appendix D. Finally, the author wishes to acknowledge the technical assistance of The Scientex Corporation in the preparation of this report.

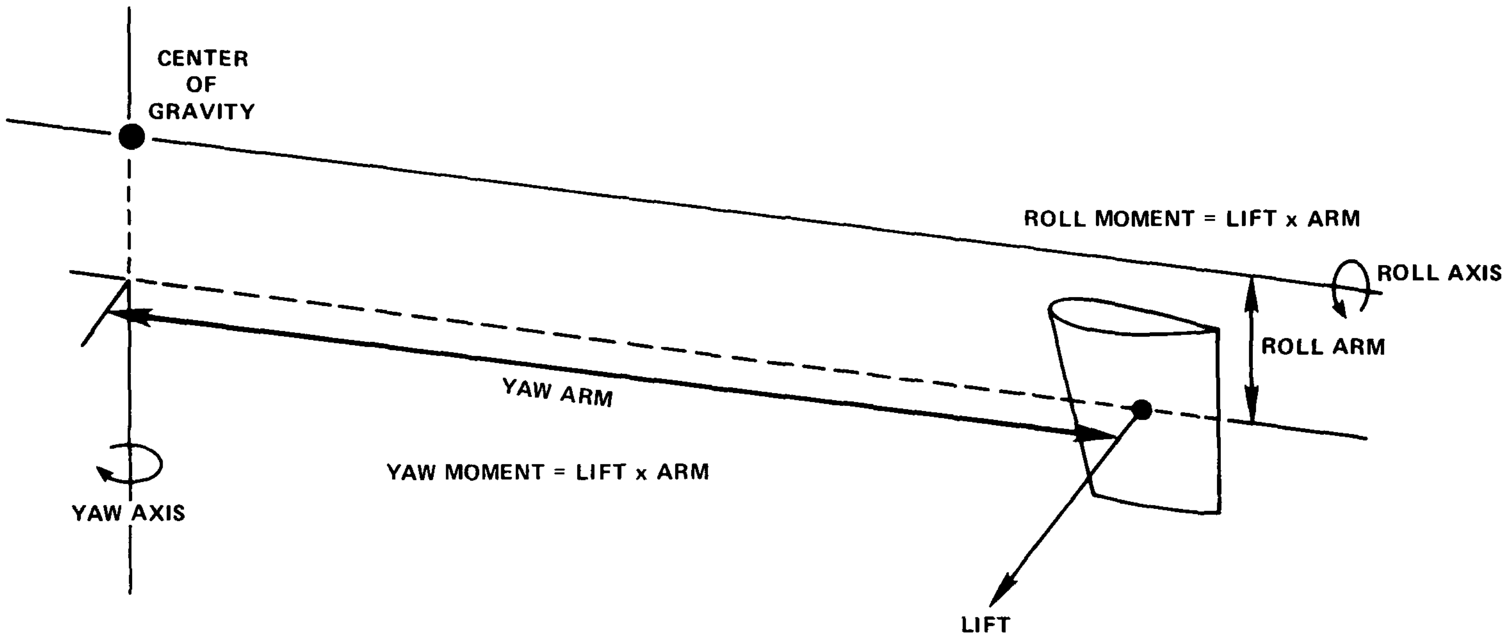


Figure 1 - Rudder Moments and Ship Responses

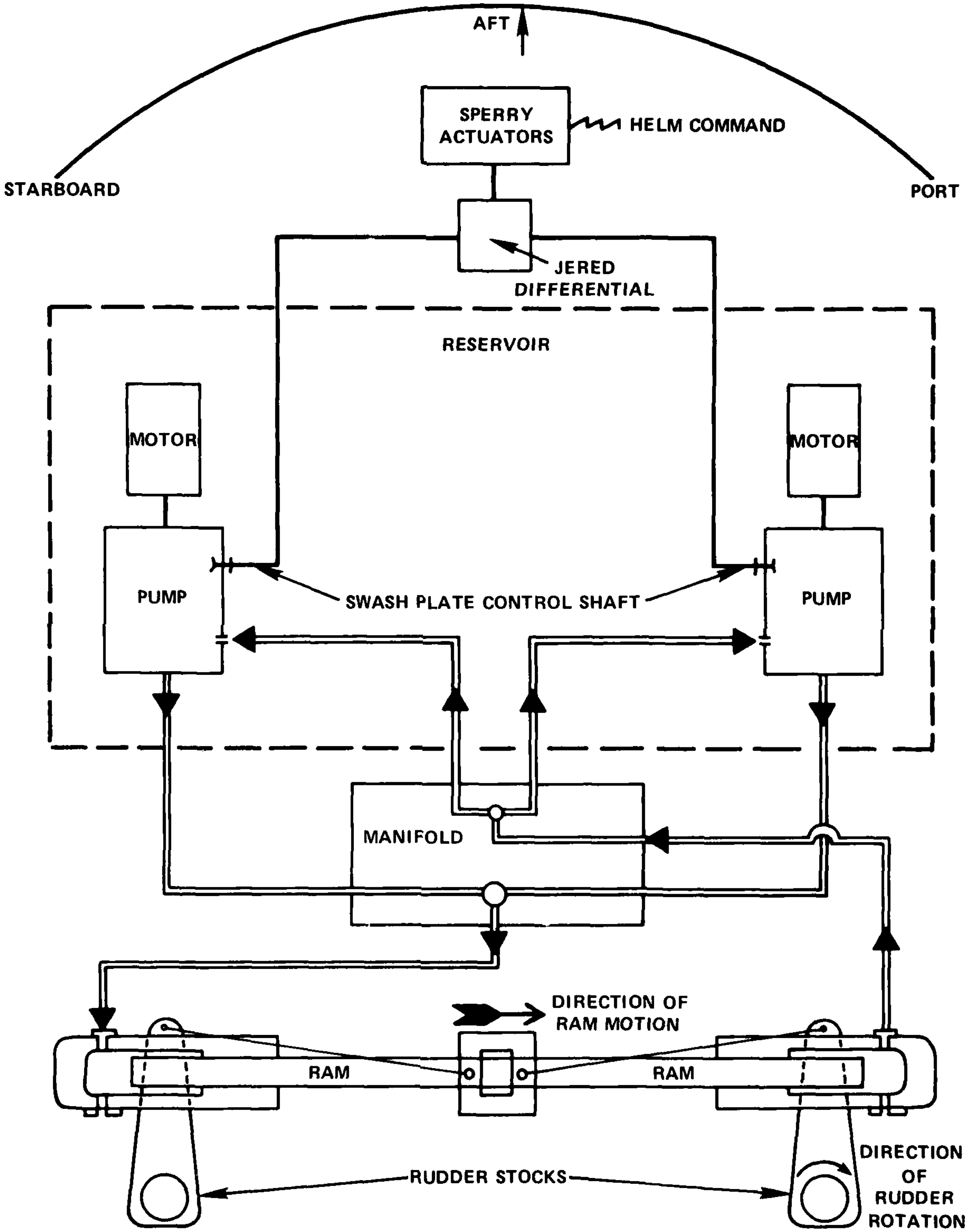


Figure 2 - Simplified Rudder System: Hydraulics + Mechanical Rams and Rudder Stocks (2 Pump Operation)

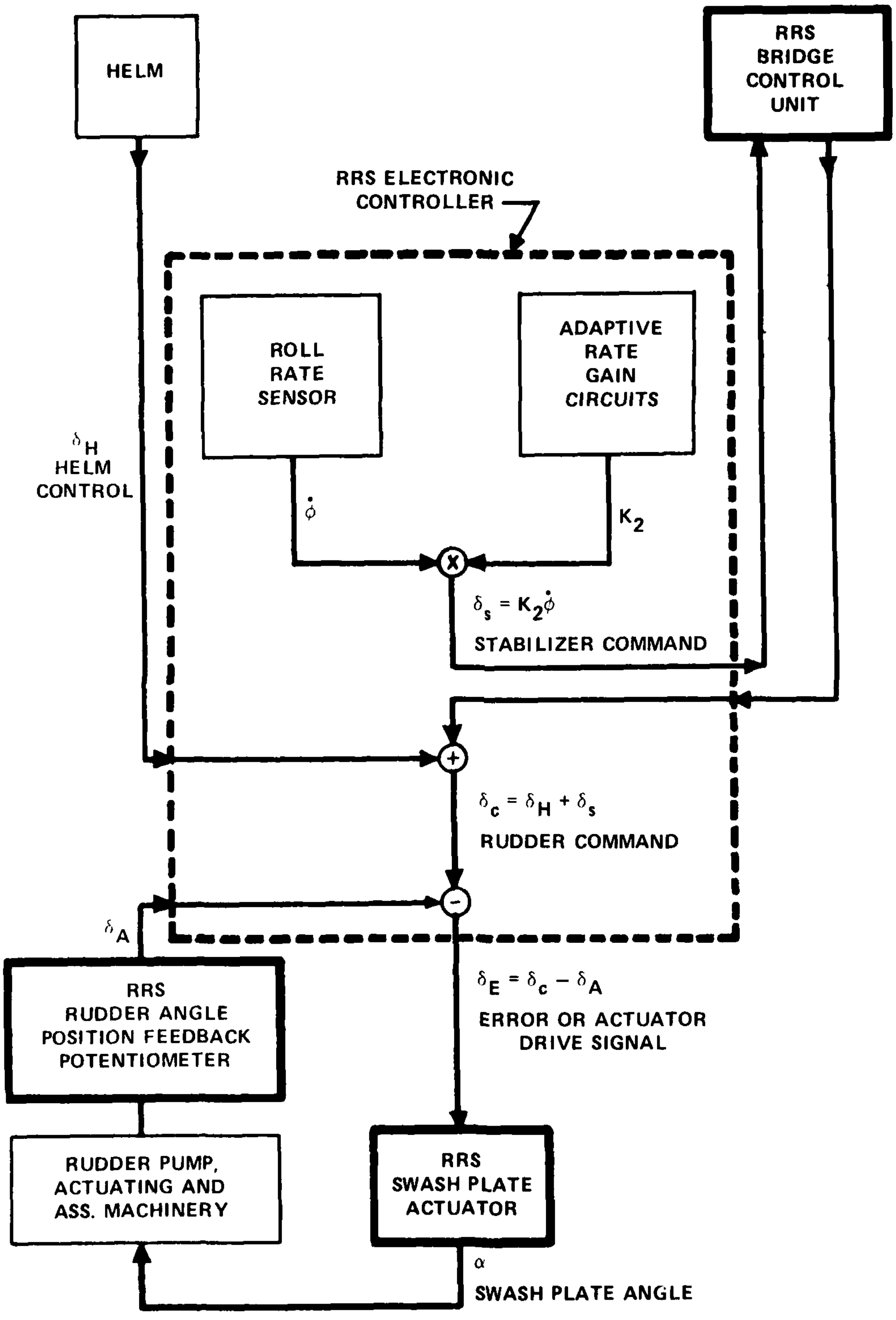


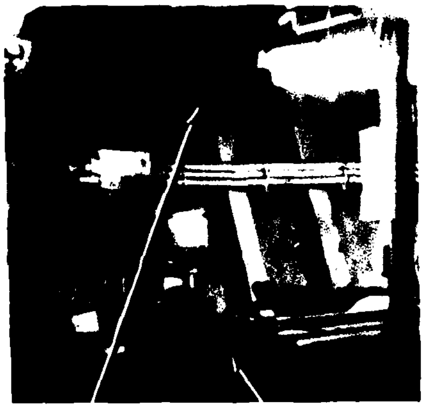
Figure 3 - Simplified RRS Electronic System Diagram

Figure 4 - RRS Components and Their Location

PRECEDING PAGE BLANK-NOT FILMED

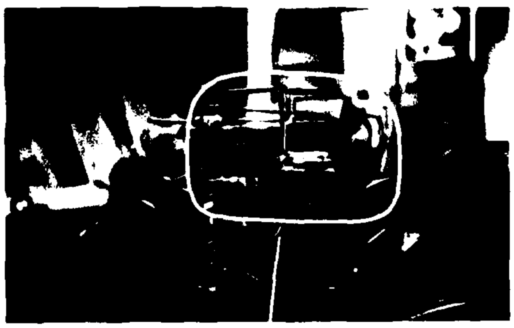
BRIDGE CONTROL UNIT

RRS SYSTEM PICTURE
JARVIS

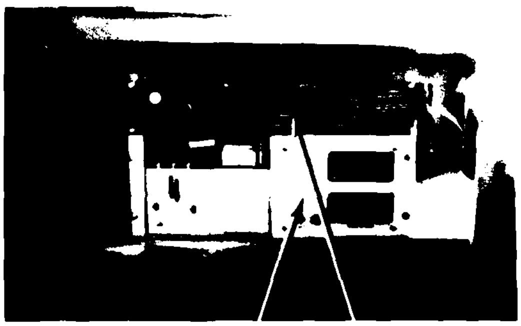


CONTROLLER AND DIFFERENTIAL

RRS SYSTEM PICTURE
MELLON

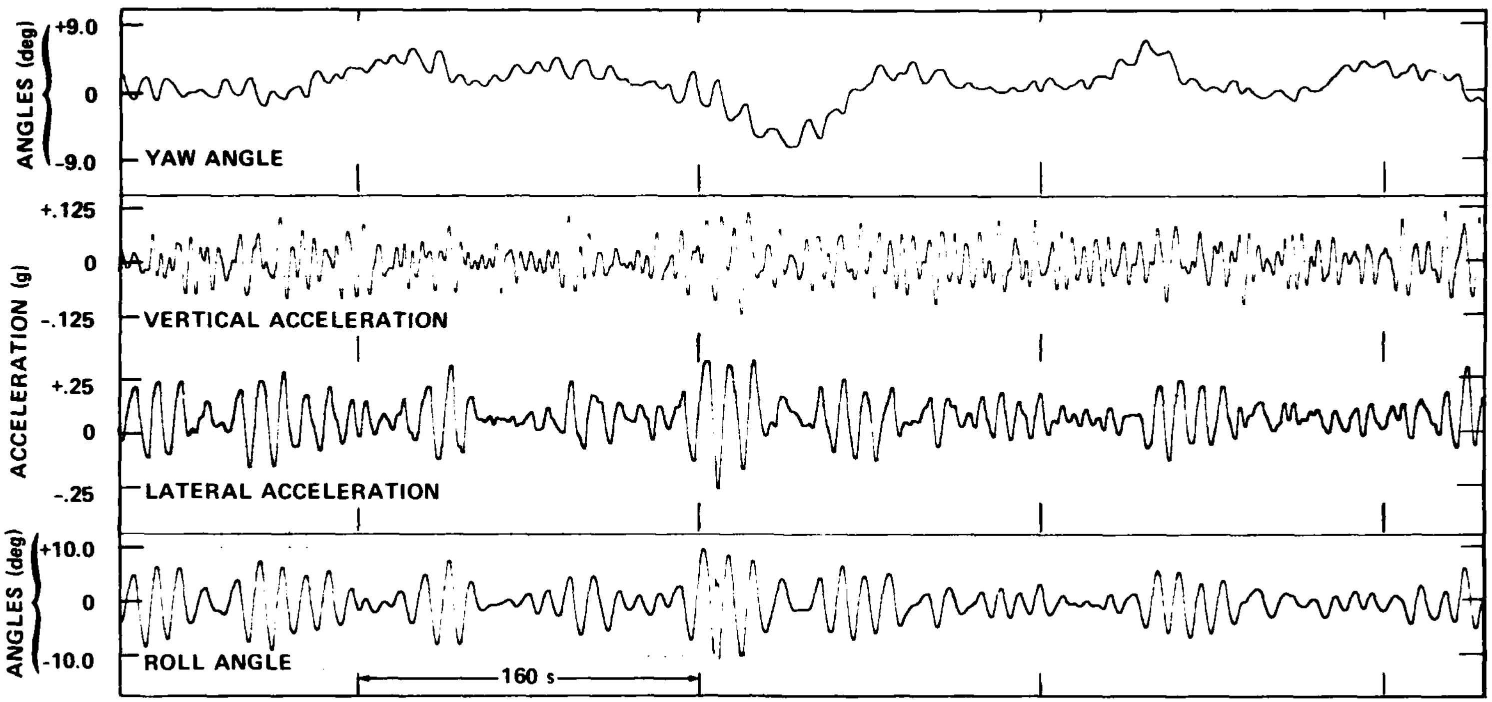


ACTUATOR



ROLL AND PITCH, RMS AND PEAK METER
JARVIS

Figure 5 - RRS System Components as Installed Aboard USCGC's MELLON and JARVIS



RUN 27 – UNSTABILIZED JARVIS RESPONSES IN 6–8 ft QUARTERING SEAS – 13.5 knots

Figure 6 - Typical Ship Motion Conditions that Severely Curtail Unrestricted Crew Movement (Walking) Without Hanging on to Retain Footing

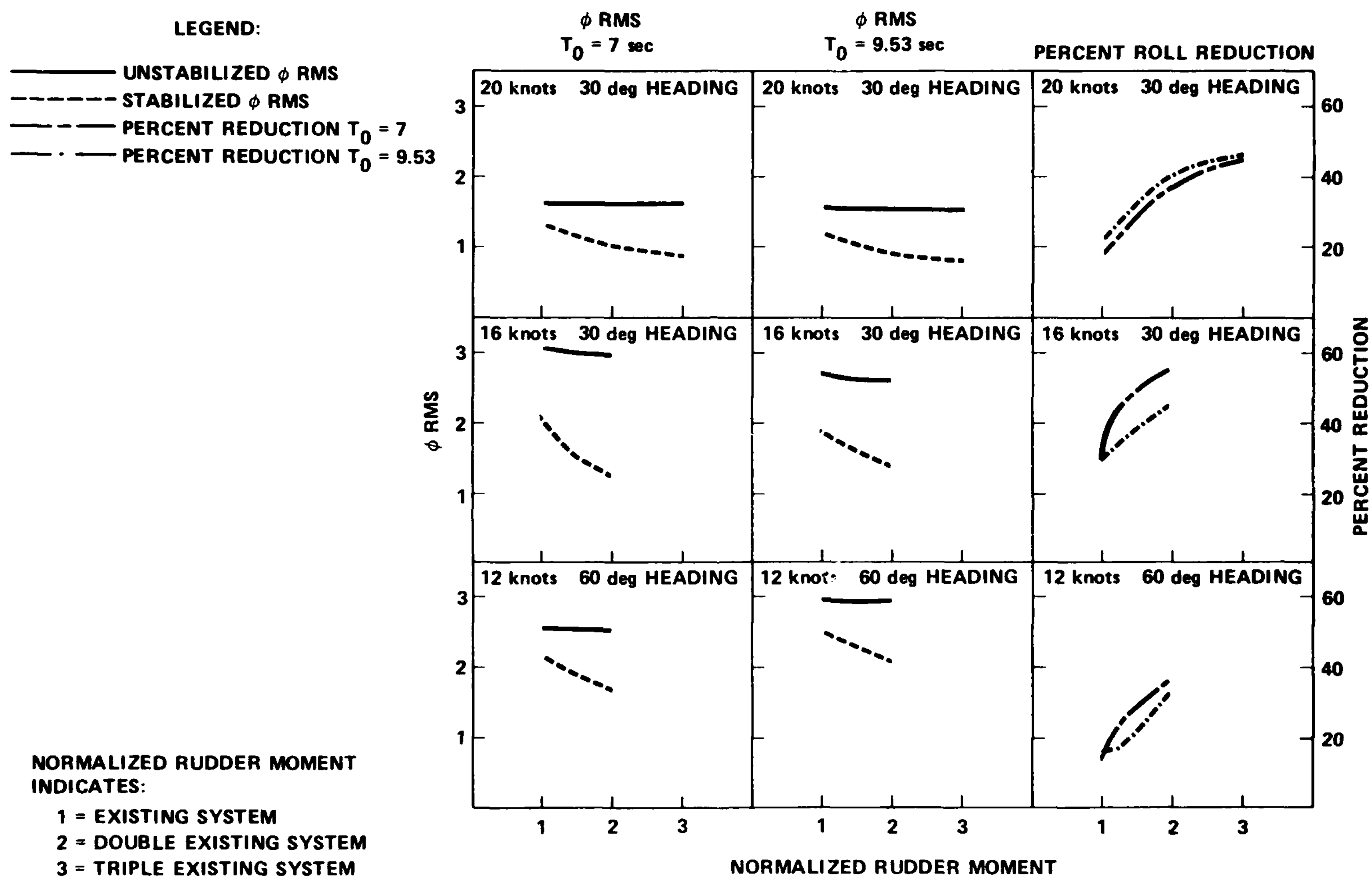


Figure 7 - Effect of Increasing Rudder Moment on Roll Reduction

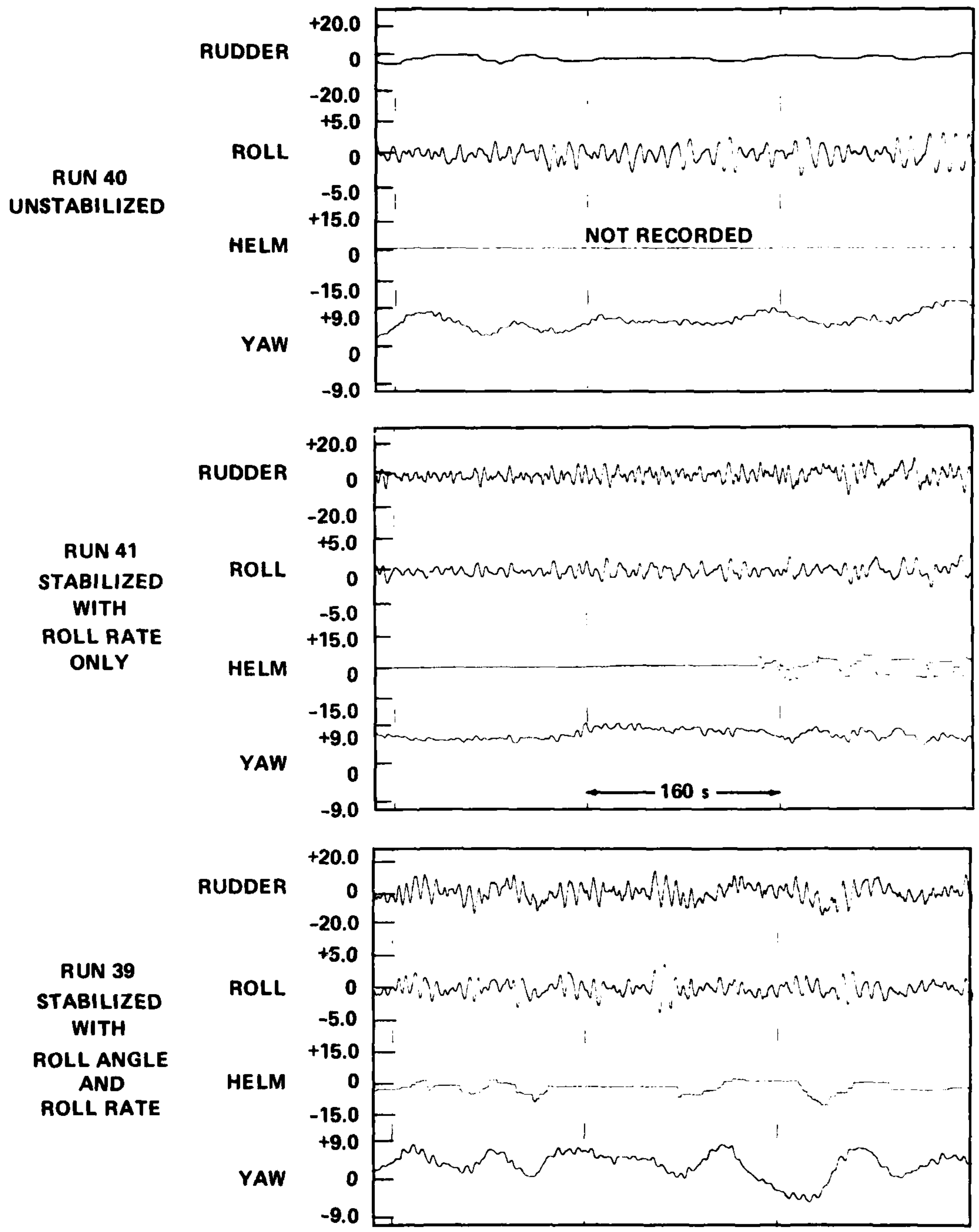


Figure 8 - Sample Time Histories of JARVIS Responses in 10 to 12 Feet Bow Seas at 13.8 Knots with Various Control Laws

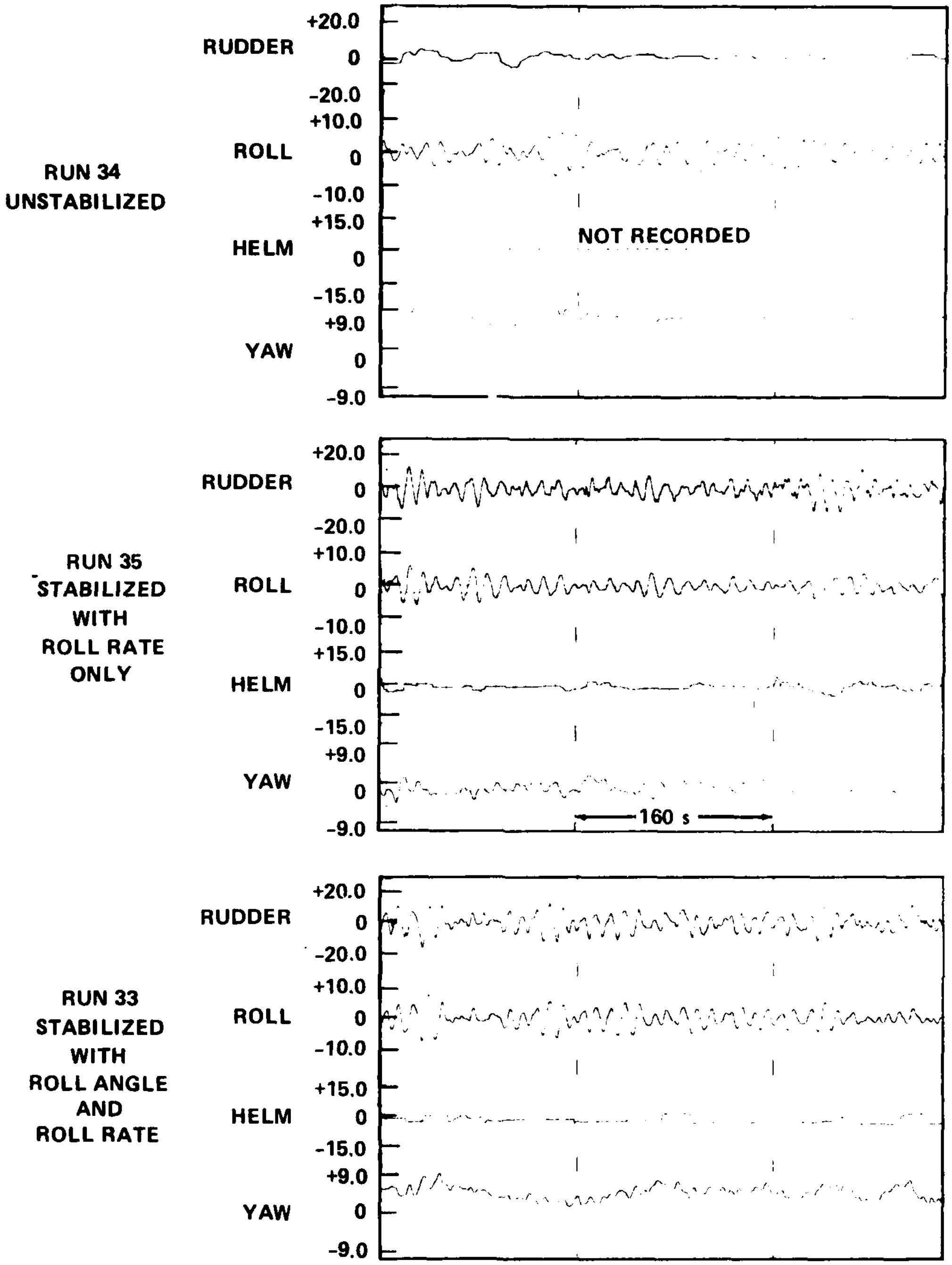


Figure 9 - Sample Time Histories of JARVIS Responses in 8 to 10 Feet Beam Seas at 14.6 Knots with Various Control Laws

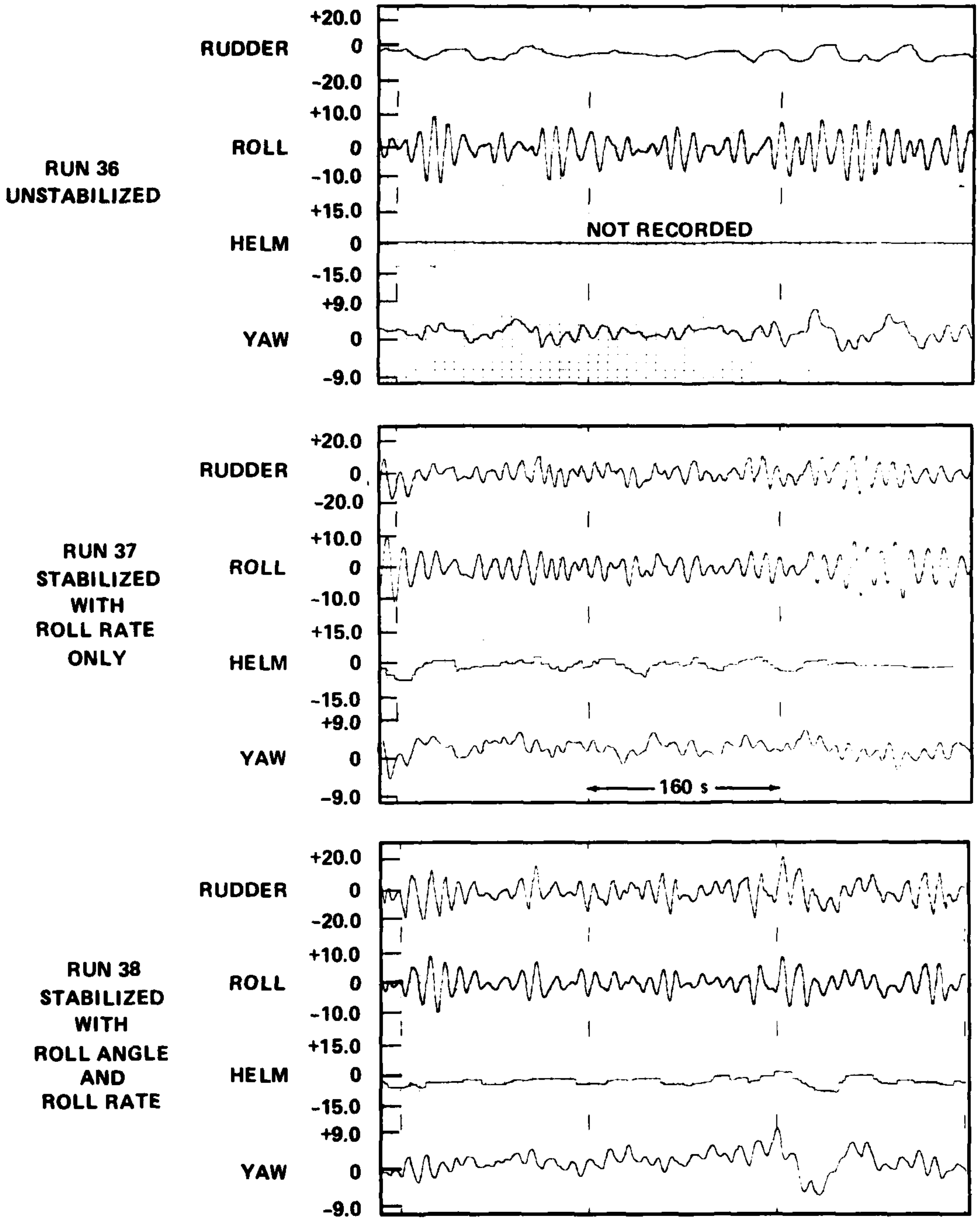


Figure 10 - Sample Time Histories of JARVIS Responses in 12 to 15 Feet Quartering Seas at 14.4 Knots with Various Control Laws

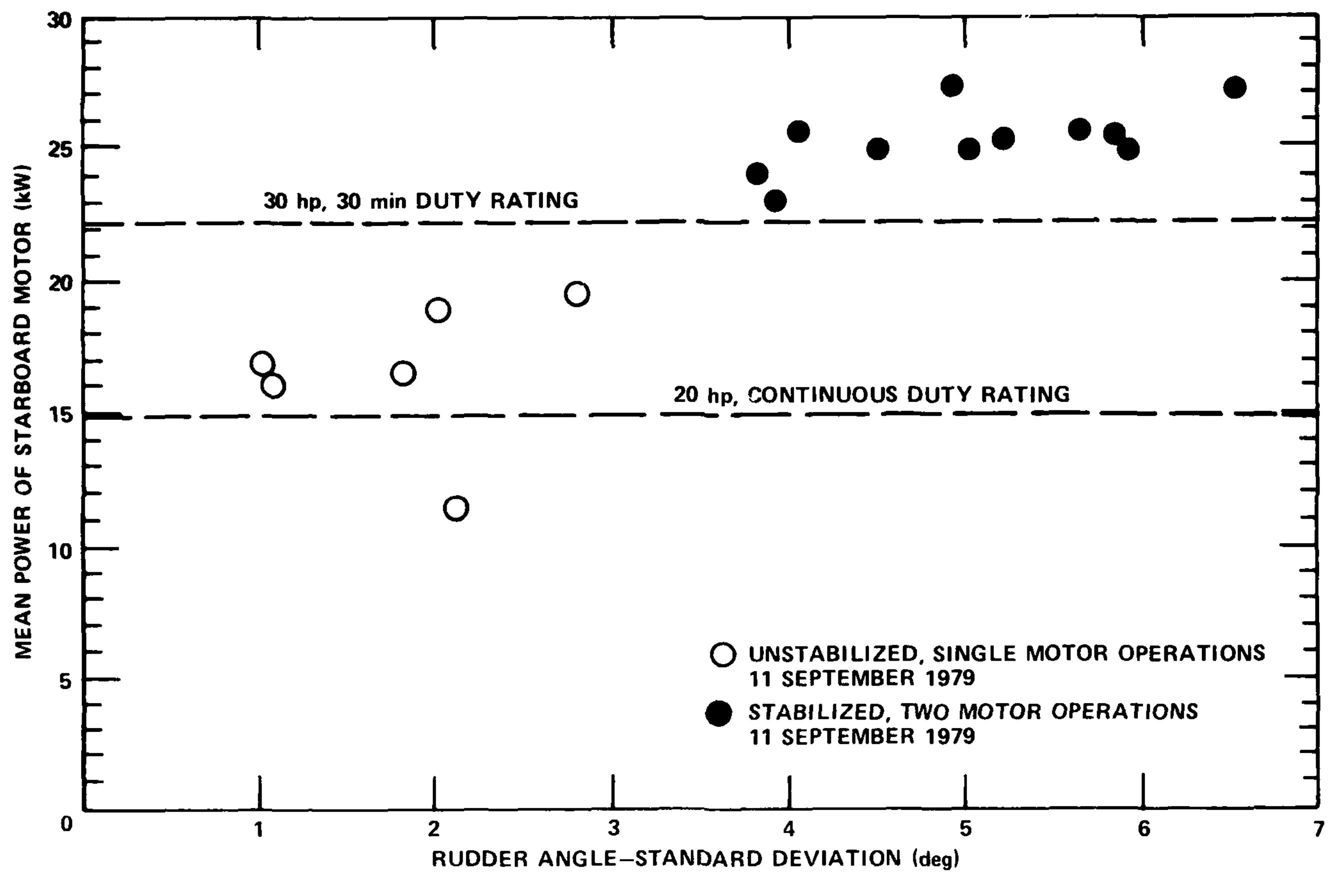


Figure 11 - Variation of Steering Motor Mean Power with Rudder Activity

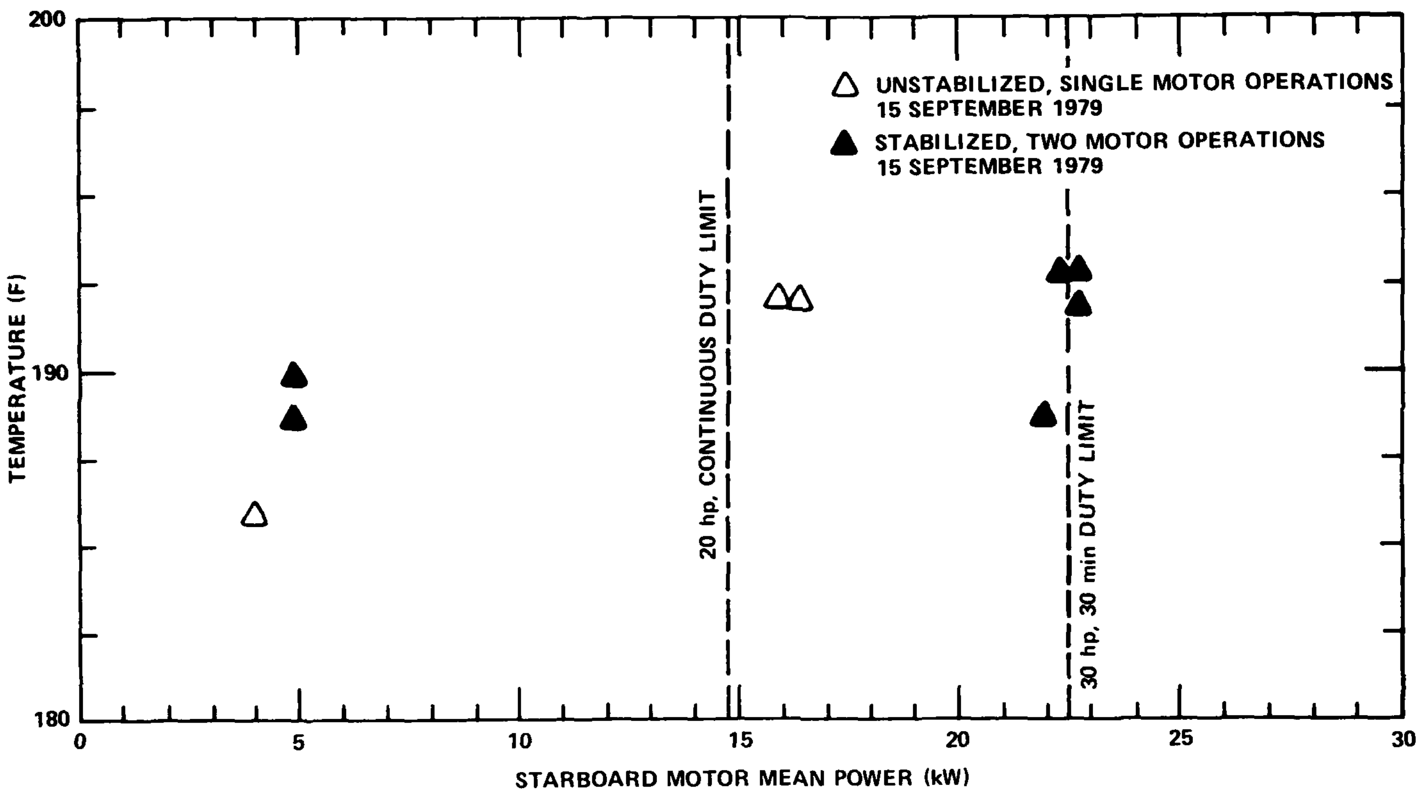


Figure 12 - Variation of Steering Motor Temperature with Mean Power

Table 1 Summary of June 1976 MELLON Trial

Trial Conditions**					Ship And Control Responses (Significant Single Amplitude)						Ship Steering And Control Responses (Standard Deviation)		
Trial Run Number	Run Length (Minutes)	Stabilizing Condition*	Observed Heading Relative To Waves	Observed Wave Height (Feet)	Roll Degrees	Roll Percent Reduction	Roll Rate Degrees/Second	Roll Rate Percent Reduction	Roll Control Signal $K_2\dot{\phi}$	Pitch Degrees	Yaw Degrees	Helm Angle Degrees	Rudder Angle Degrees
1	8	U	Beam	5-7	5.34		3.30			1.13	0.71	1.70	1.70
2	20	U	"	"	5.36		3.34			1.19	1.41	2.81	2.33
3	20	C	"	"	3.56	33.6/32.5	2.17	36.4/42.6	4.11	1.09	1.38	2.40	3.73
4	20	U	"	"	5.79		3.78			1.17	1.25	1.82	1.93
5	15	A	"	"	3.55	38.7	2.21	41.5	4.60	1.08	1.75	1.45	3.15
6	22	U	"	6-8	7.18		4.50			1.18	1.10	1.54	1.67
7	18	C	"	"	4.33	39.7/37.3	2.68	40.4/38.7	5.05	1.23	0.98	3.21	4.38
8	20	U	"	"	6.91		4.37			1.25	1.72	3.41	3.50
9	40	U	Quartering	5-8	4.29		1.87			1.05	1.53	0.83	0.90
10	40	C	"	"	3.36	21.7/26.8	1.12	40.1/47.9	3.03	0.76	2.21	1.48	2.44
11	40	U	"	"	4.59		2.15			1.08	2.40	1.05	1.12

*U = Unstabilized
C = Stabilized with a constant K_2 value
A = Stabilized with an adaptive K_2 value

**Ship speed was 15 knots for all runs listed.

Table 2 Summary of September 1979 JARVIS Trial

Trial Conditions							Lateral Responses (Significant Single Amplitude)				Vertical Responses (Significant Single Amplitude)		Ship Steering And Control Responses (Standard Deviation)		
Trial Series Number	Trial Run Number	Run Length (Minutes)	Stabilizing Condition*	Observed Heading Relative To Waves	Observed Wave Height (Feet)	Measured Ship Speed (Knots)	Roll Angle Degrees	Roll Angle Percent Reduction	Lateral Acceleration g's	Lateral Acceleration Percent Reduction	Pitch Angle Degrees	Vertical Acceleration g's	Yaw Angle Degrees	Helm Angle Degrees	Rudder Angle Degrees
Stabilization With Roll Rate Only															
[illegible]	[illegible]	29	U	Head/Bow	5-7	[illegible]	2.03		0.063		1.57	0.076	-	-	1.8
[illegible]	2	30	S	"	"	"	1.59	22.7/21.1	0.056	11.1/23.3	1.39	0.074	-	3.3	2.9
[illegible]	[illegible]	[illegible]	U	"	"	"	1.99		0.073		1.27	0.068	-	-	1.1
[illegible]	24	[illegible]	U	Bow	6-7	14.0	2.22		0.082		1.84	0.102	1.30	-	0.4
[illegible]	26	20	S	"	"	"	1.28	42.3	0.065	20.7	1.56	0.086	0.84	0.6	0.5
[illegible]	21	15	[illegible]	Bow	8-12	13.6	3.77		0.123		1.93	0.105	-	-	3.1
[illegible]	22	14	S	"	"	"	3.02	23.9	0.107	13.0	1.72	0.100	-	1.1	4.3
8	30	25	U	"	"	14.8	7.95		0.209		1.79	0.112	2.06	-	2.5
8	[illegible]	30	S	"	"	"	4.89	38.5	0.166	20.6	1.72	0.118	1.74	0.8	4.6
[illegible]	40	30	U	"	10-12	13.6	2.60		0.093		1.98	0.107	1.66	-	1.3
[illegible]	[illegible]	[illegible]	S	"	"	13.8	1.93	25.8	0.089	6.5	2.06	0.112	1.23	2.6	4.4
4	18	33	U	Beam	7-8	13.9	4.83		0.140		1.31	0.087	-	-	2.1
4	20	30	S	"	"	14.6	2.45	49.3	0.099	29.3	1.57	0.095	-	1.3	4.1
9	34	23	U	Beam/Bow	8-10	14.5	6.35		0.172		1.76	0.122	1.74	-	1.8
9	35	23	S	"	"	14.6	4.36	31.3	0.143	16.9	1.70	0.116	4.12	1.6	4.8
2	4	30	U	Quartering	6-7	13.5	3.14		0.103		1.11	0.078	-	-	1.1
2	5	30	S	"	"	13.9	2.25	28.3	0.090	12.6	1.11	0.074	-	0.8	3.8
[illegible]	27	20	U	"	6-8	13.5	6.53		0.177		1.48	0.082	2.2	-	0.2
7	28	20	S	"	"	"	4.67	28.5	0.148	16.4	1.49	0.089	[illegible]	0.9	0.9
[illegible]	11	30	U	"	"	15.2	3.07		0.099		1.08	0.078	-	-	1.0
3	12	30	S	"	"	"	2.23	27.4	0.089	10.1	1.11	0.078	-	2.1	4.0
10	36	30	U	"	12-15	14.9	7.81		0.186		1.66	0.078	1.77	-	2.8
10	37	30	S	"	"	"	7.16	8.3	0.189	-1.6	2.02	0.077	2.01	1.9	5.1
Stabilization With Roll Angle and Roll Rate															
7	29	20	S	Quartering	6-8	13.5	4.75	27.3	0.149	20.5	1.72	0.114	1.87	1.9	1.1
10	38	30	S		12-15	14.4	6.20	20.6	0.159	14.5	2.15	0.082	2.69	1.7	6.4
9	33	30	S	Beam	8-10	14.2	4.30	32.3	0.133	22.7	1.78	0.116	2.73	1.8	5.5
11	39	30	S	Bow	10-12	13.6	2.54	2.3	0.097	-4.3	2.48	0.125	3.18	3.5	6.3
6	25	40	S	"	6-7	14.0	2.31	-4.1	0.079	3.7	1.71	0.092	3.38	2.9	1.1

*U = Unstabilized
S = Stabilized

APPENDIX A
DETAILS OF RRS PROTOTYPE DEVELOPMENT

The USCGC HAMILTON was selected for prototype development after ship/helicopter interface trials in April 1975 established the desirability of roll stabilization. Since DTNSRDC wanted to validate the results of the recently completed RRS feasibility study, a joint USN/USCG effort was initiated to develop an RRS system for HAMILTON.

During the HAMILTON trials in October 1975, rudder roll moment capacity was determined by oscillating the ship's rudder and measuring the induced roll. Various rudder motion amplitudes and frequencies were obtained using a sine wave generator to simulate the rudder command. The results of these forced roll experiments are shown in Figure A1 for ship speeds of 15, 20 and 25 knots. Responses indicate that adequate roll moment is available for roll reduction if rudder excursion amplitude and phasing are correctly applied. During the trials, rudder excursion was maintained at the maximum amplitude obtainable without commanding the rudder to exceed a rate of 4.66 degrees per second (the maximum design value for the steering system). While these results were promising, a lag between the rudder command signal and the actual rudder response was discovered. This lag was unacceptable from the point of view of roll control, though satisfactory for steering. Its source was traced to the JERED differential in the ship's swash plate drive system, noted in Figure 2. The differential was designed to limit rudder response for small helm signals, as shown in Figure A2. This diagram which shows rudder rate versus error signal (solid line) reveals that, for error signals less than 7 degrees, the available rudder rate is less than maximum. However, to achieve the maximum benefit from a roll reduction system, the rudder rate should reach the available maximum as rapidly as possible on demand. Hence, it was necessary to modify the rudder control.

MOD 1

The MOD 1 stage of the prototype development consisted of building a swash plate drive system for the steering pumps that imposed the rudder command on the pumps without the JERED differential induced lags of the existing steering system. The resulting rudder response is shown by the

dotted line in Figure A2 which reveals that the rudder lag was minimized by allowing the steering pumps to produce maximum rudder rate for relatively small rudder commands.

Dockside trials were then conducted aboard HAMILTON in January 1976. The results of these trials indicated that the rudder could be actuated with the required phasing. However, prior to installation of the prototype system, a review with USCG officials established the safety specification that, if the RRS swash plate drive system should fail or be shut down for any reason, conventional steering must be automatically restored. This system requirement led to the next modification.

MOD 2

To meet the newly imposed safety requirement, a fail-safe clutch was developed and integrated into the RRS system. To evaluate the effectiveness of the modified system in significant seas, the RRS system was installed on MELLON (of the HAMILTON Class), which was scheduled for a 72-day Alaskan patrol. The system was energized during transit from Honolulu to Kodiak in May 1976, which included operations in high seas (35 foot or 10.7 meter significant wave height). Although the RRS system worked, two problems were discovered. The fail-safe clutch mechanism had reintroduced a lag into the system response and, at large roll angles, the steering system rate-saturated, causing the rudder response to fall behind to the point where the roll was actually being increased. To solve these problems, additional system modifications were necessary.

MOD 3

The lag induced by the fail-safe clutch was found to be mechanical in nature and was minimized by improving the linkage between the clutch and the swash plate control shaft. The phenomenon of rate-saturation was caused by the inability of the steering system to move the rudder quickly enough to track the stabilizer command signal. Because the stabilizer command signal, as shown in Figure 3, consists of the product of the roll rate, $\dot{\phi}$, and a control constant, K_2, rate-saturation can be either induced or negated by choosing larger or smaller values of K_2 for a given $\dot{\phi}$. Calculations based on the natural roll period of the vessel, the

maximum rudder rate available and statistical properties of the roll time history determined that an average rudder excursion of just over 5 degrees provides an optimum tradeoff between the large rudder angles necessary for large rudder moments and the alleviation of rate-saturation for all but the largest values of roll rate during any arbitrary time period.

Figure A3 summarizes the effect of K_2 on the rudder response for a simplistic sinusoidal roll rate signal with a period equal to the natural period of the vessel. When the product of K_2 and $\dot{\phi}$ is small, as shown in Figure A3a, the rudder response is in phase with $\dot{\phi}$ as desired for rate control. However, its amplitude is too small to produce the necessary roll moment. Increasing K_2 until a rudder excursion of approximately 5 degrees is obtained yields a rudder response which is larger and still in phase with $\dot{\phi}$ as shown in Figure A3b. Increasing K_2 further will result in rate-saturation as shown in Figure A3c. While the actual rudder response yields desirable large excursions, these excursions are not in phase with the commanded response. This change in phase produces roll destabilization and cannot be allowed to occur.

The addition of an adaptive control circuit which varies the value of K_2 was required. This circuit was designed to provide a K_2 value based on a running average of the roll rate previously experienced. The time constants for this circuit were calculated using the statistical characteristics of the vessel motion and the desired rudder excursion. With the addition of this circuit, the RRS system became capable of providing optimum roll stabilization, within current design goals and constraints, over a wide range of $\dot{\phi}$ values.

The effectiveness of the system was evaluated aboard MELLON en route from Seattle to Honolulu in June 1976. This evaluation revealed that the RRS system was feasible for short duty cycles. Figure A4 shows that, in typical back-to-back measurements of unstabilized and stabilized ship roll in beam seas, the significant roll was reduced by about 35 percent. In addition, the beating phenomenon was significantly reduced when the ship was stabilized. Figure A5 displays the yaw measurements for the same time period and reveals that yaw did not change significantly. Figure A6 shows the rudder angle for the same time period and indicates that both the frequency and amplitude of the rudder action definitely

increased. Finally, in Figure A7, the helm command is shown, again for the same time period. The data shown in this figure indicate that the helm command did not change significantly when the RRS system was activated. However, experience in a variety of seas at various headings revealed that the helm action was generally reduced, indicating that the steering control task was improved by the RRS system. Discussions with several helmsmen substantiated this conclusion.

MOD 4

The RRS system, as designed and developed up to this point, was a working research prototype. Though the feasibility of the system had been demonstrated, a preproduction prototype, suitable for operator evaluation, still needed to be constructed.

Two preproduction prototypes were built in 1977. Originally, one was to be installed on HAMILTON for East Coast deployments and the other on MELLON for deployments in the North Pacific and Bering Sea so that the system could be evaluated in a variety of sea conditions. Funding, however, was insufficient, and the HAMILTON installation could not be completed.

The RRS system installation was completed on MELLON prior to her departure from Honolulu to Kodiak in January 1978. Unfortunately, MELLON developed problems with her controllable pitch propellers shortly after departure, and this prevented a system evaluation. The propeller problem could not be corrected in Kodiak, so MELLON returned to Honolulu. On this return voyage, the RRS system was activated but did not appear to function properly. It was determined that a filter in the hydraulic system was clogged. In addition, a faulty electronic chip was discovered in the adaptive controller.

The RRS system was repaired, and the propeller problem was corrected prior to the Summer 1978 Alaskan patrol. Though no data were recorded during this patrol, the crew used the system continuously for 56 days (far exceeding the original design concept of a short duty cycle) and was very pleased with the roll reduction capability of the RRS system. At the end of the 56-day period, the system was shut down because one of the

two steering pump motors failed due to overheating, and the RRS system would not function properly with only one steering motor.

MOD 5

The next modifications to the RRS system were implemented during 1979 and consisted of new control circuitry and other minor improvements. Two preproduction prototype controllers were upgraded to MOD 5 and installed on MELLON and JARVIS, both cutters of the HAMILTON Class. During the time between modifications, the crew on board MELLON had completely changed so a briefing was held for the crews of both ships in which details of the system and the desired operational evaluation parameters were reviewed. JARVIS left Honolulu in September 1979 for Alaskan patrol and returned in mid-November. MELLON left Honolulu, also for Alaskan patrol, in October 1979 and returned in December. During these trials, roll stabilization data were collected for a variety of sea conditions, headings and vessel speeds.

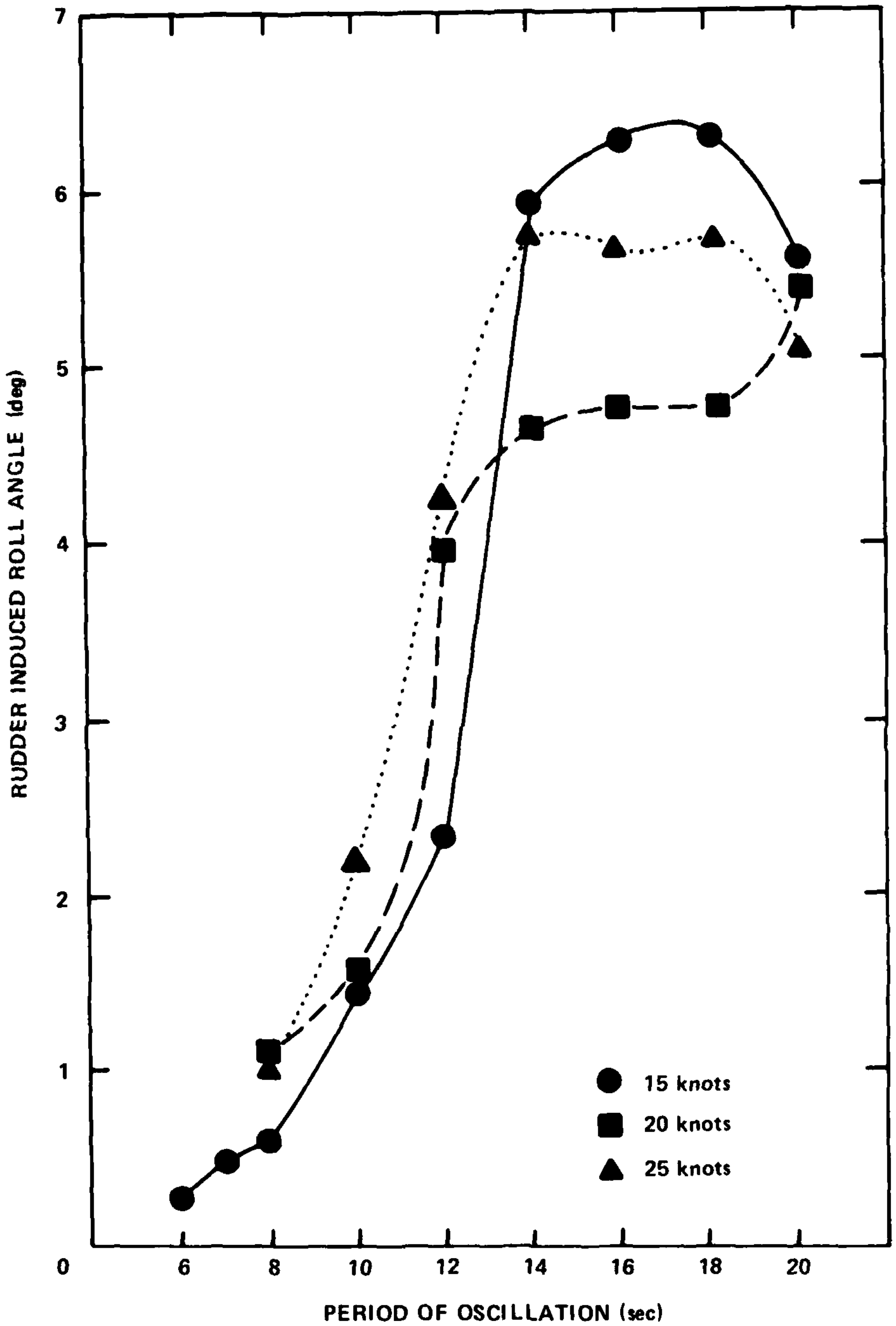


Figure A1 - Forced Roll at Various Ship Speeds

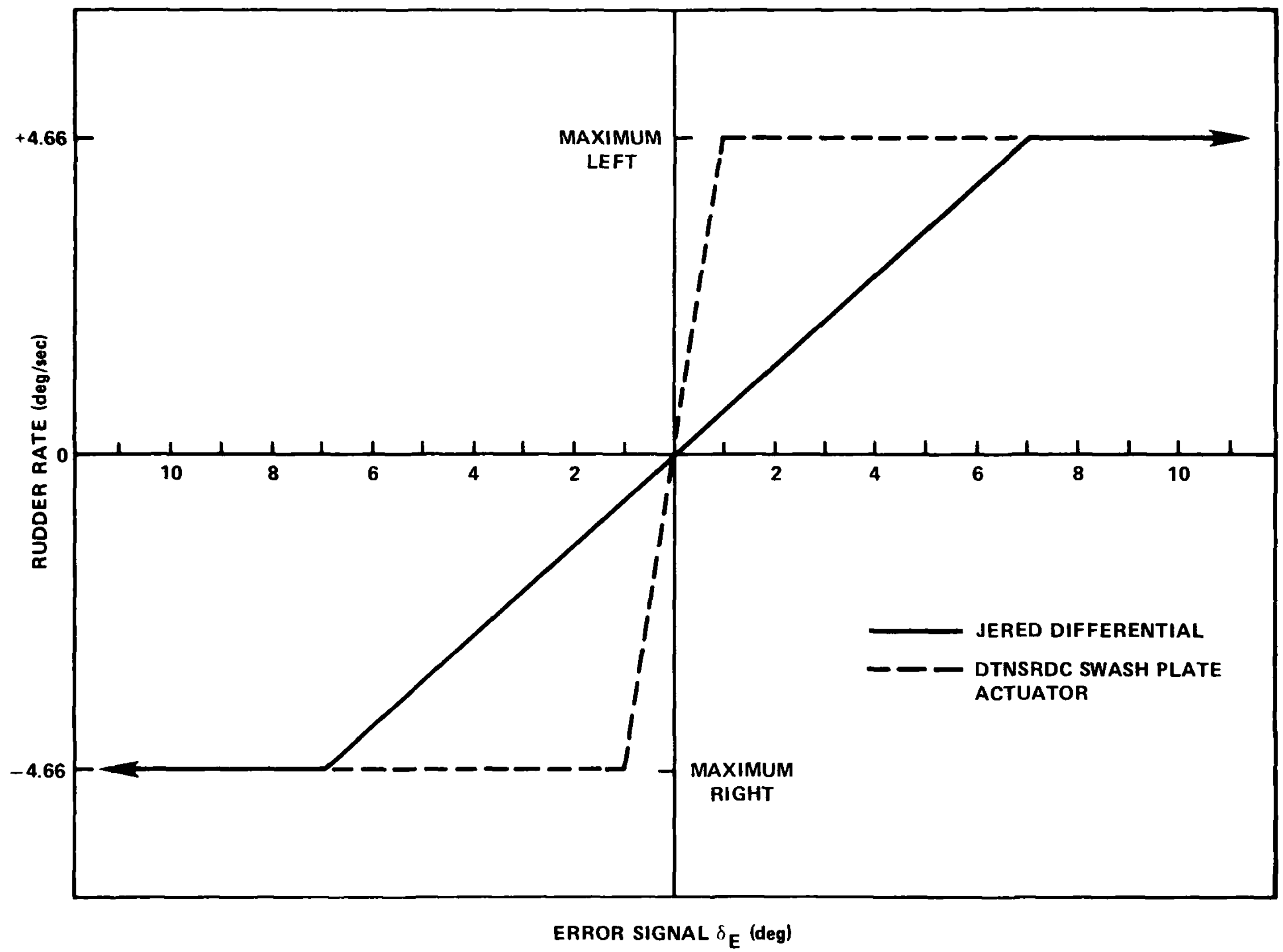


Figure A2 – Rudder Rate versus Rudder Position Error

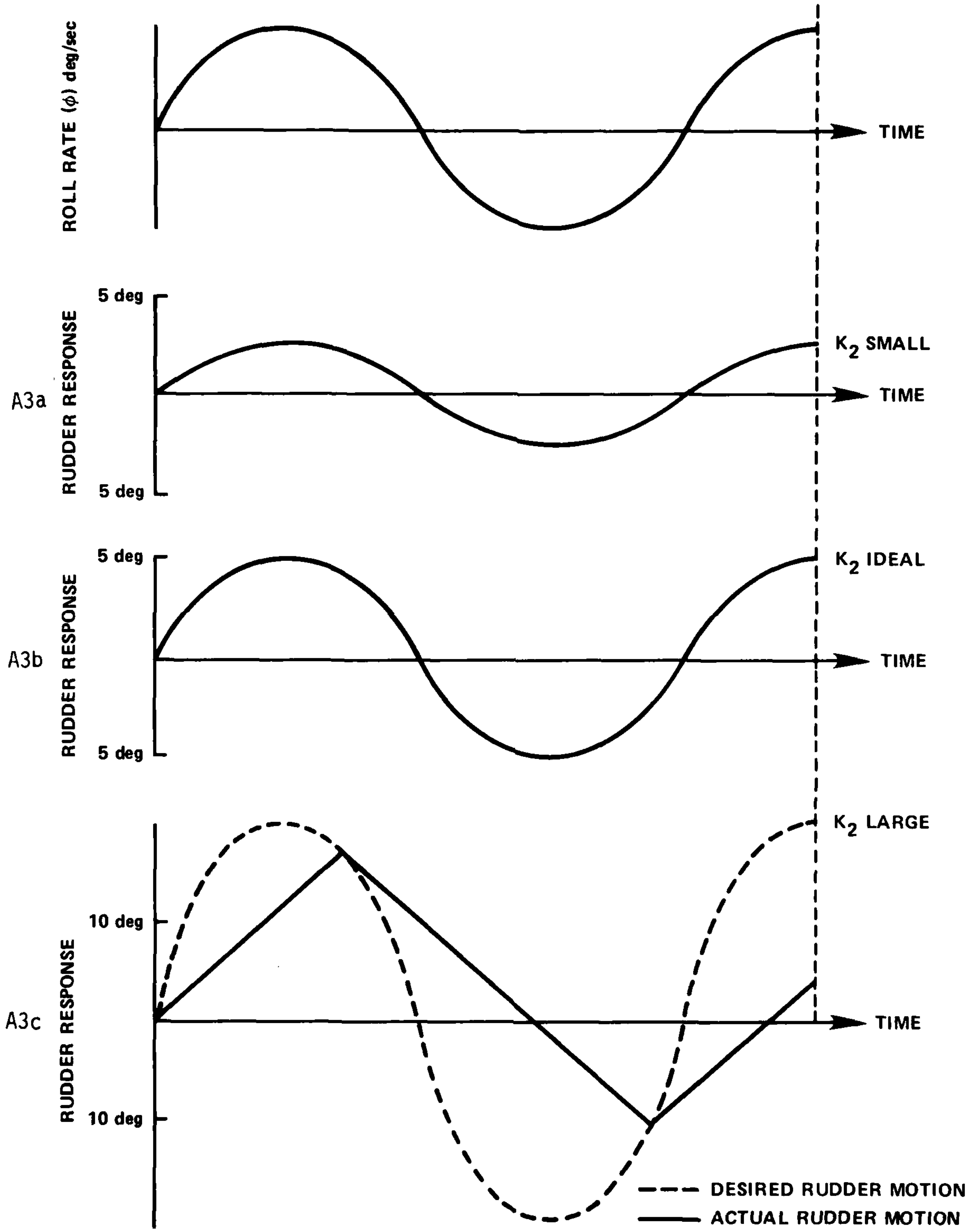


Figure A3 - Effect of K_2 on Rate Saturation

USCGC MELLON ROLL STABILIZATION TRIAL
BEAM SEAS, 5–7 FEET, 15 KNOTS

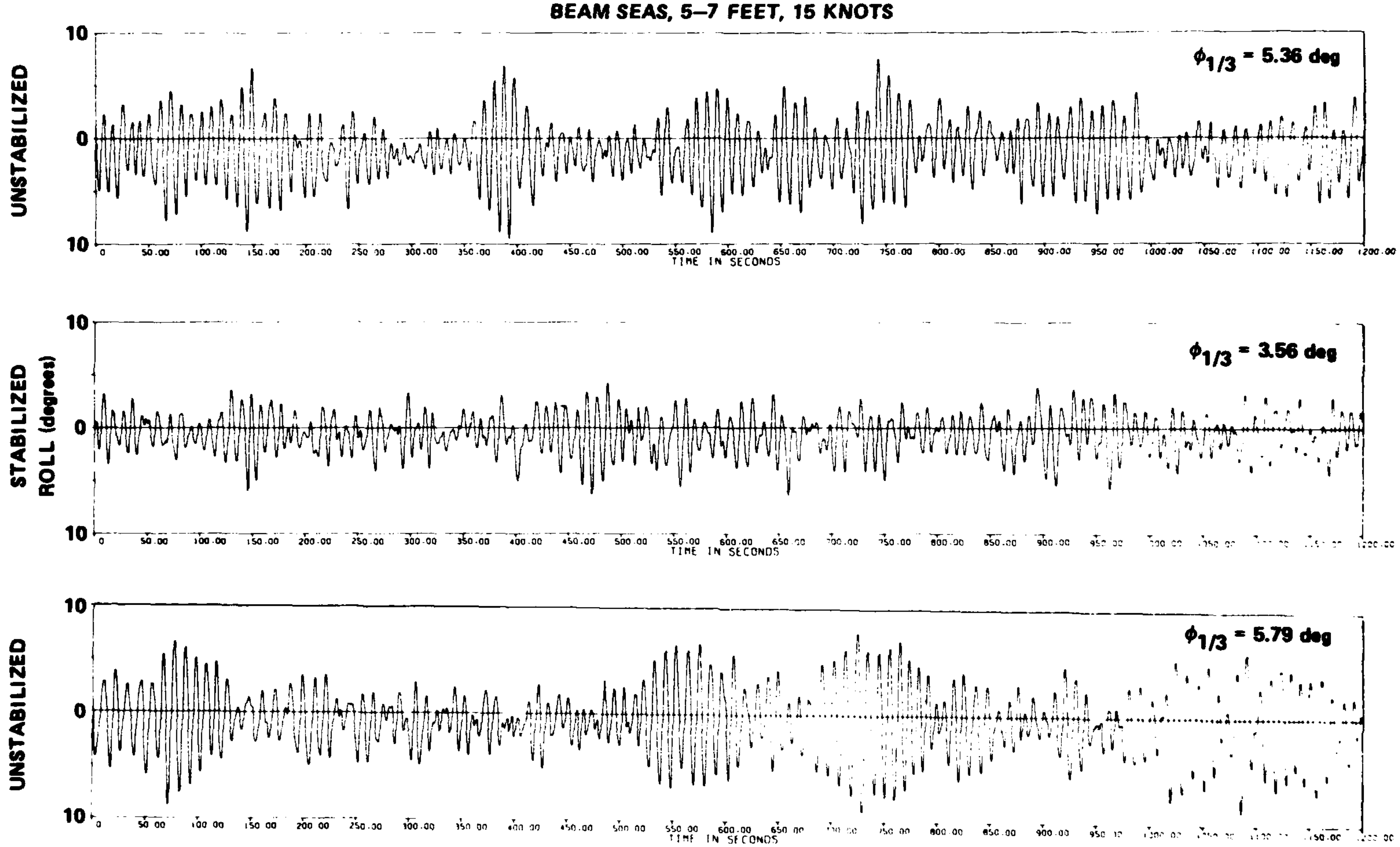


Figure A4 - Measured Roll Angle

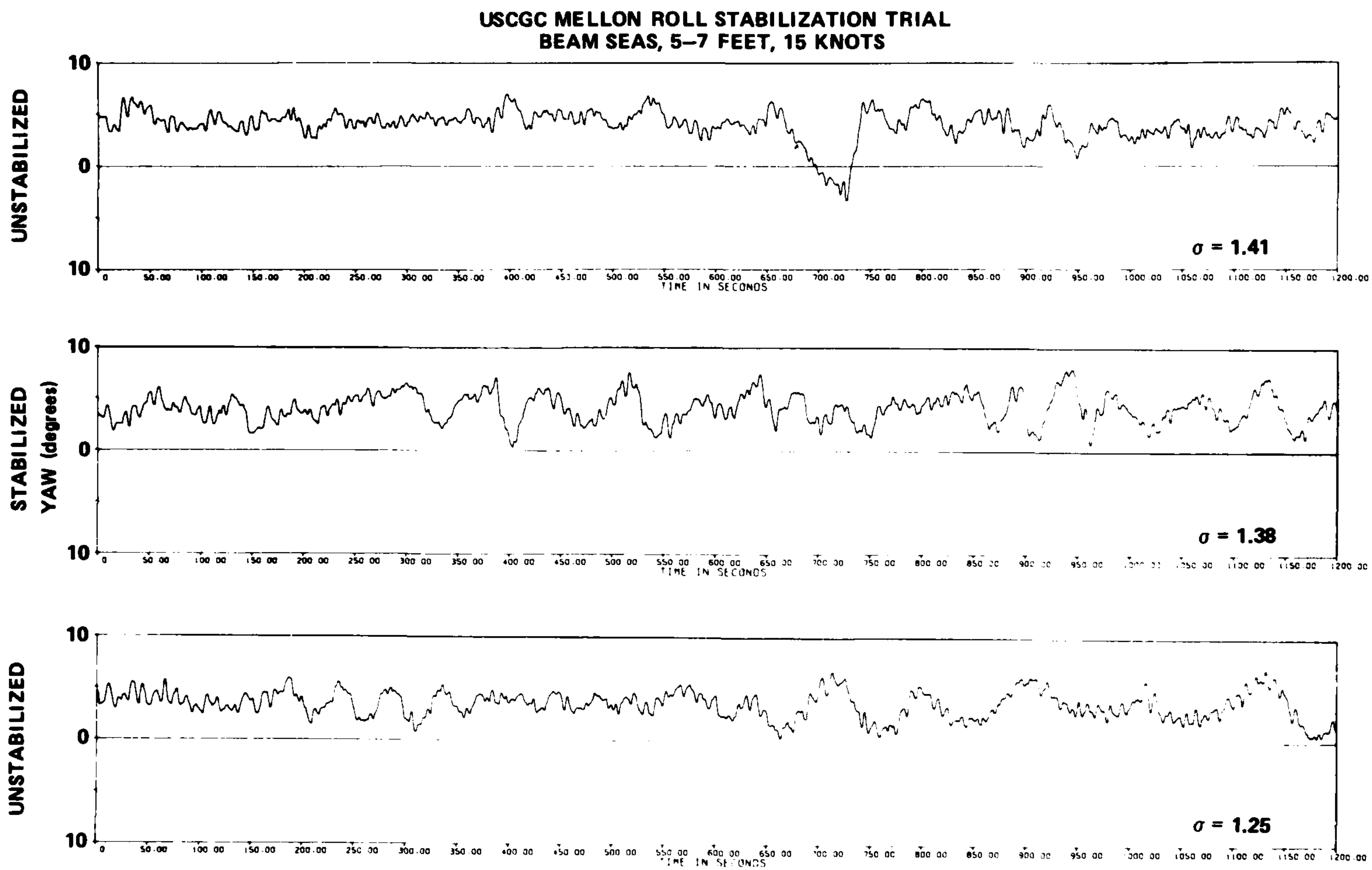


Figure A5 - Measured Yaw Angle

USCGC MELLON ROLL STABILIZATION TRIAL
BEAM SEAS, 5–7 FEET, 15 KNOTS

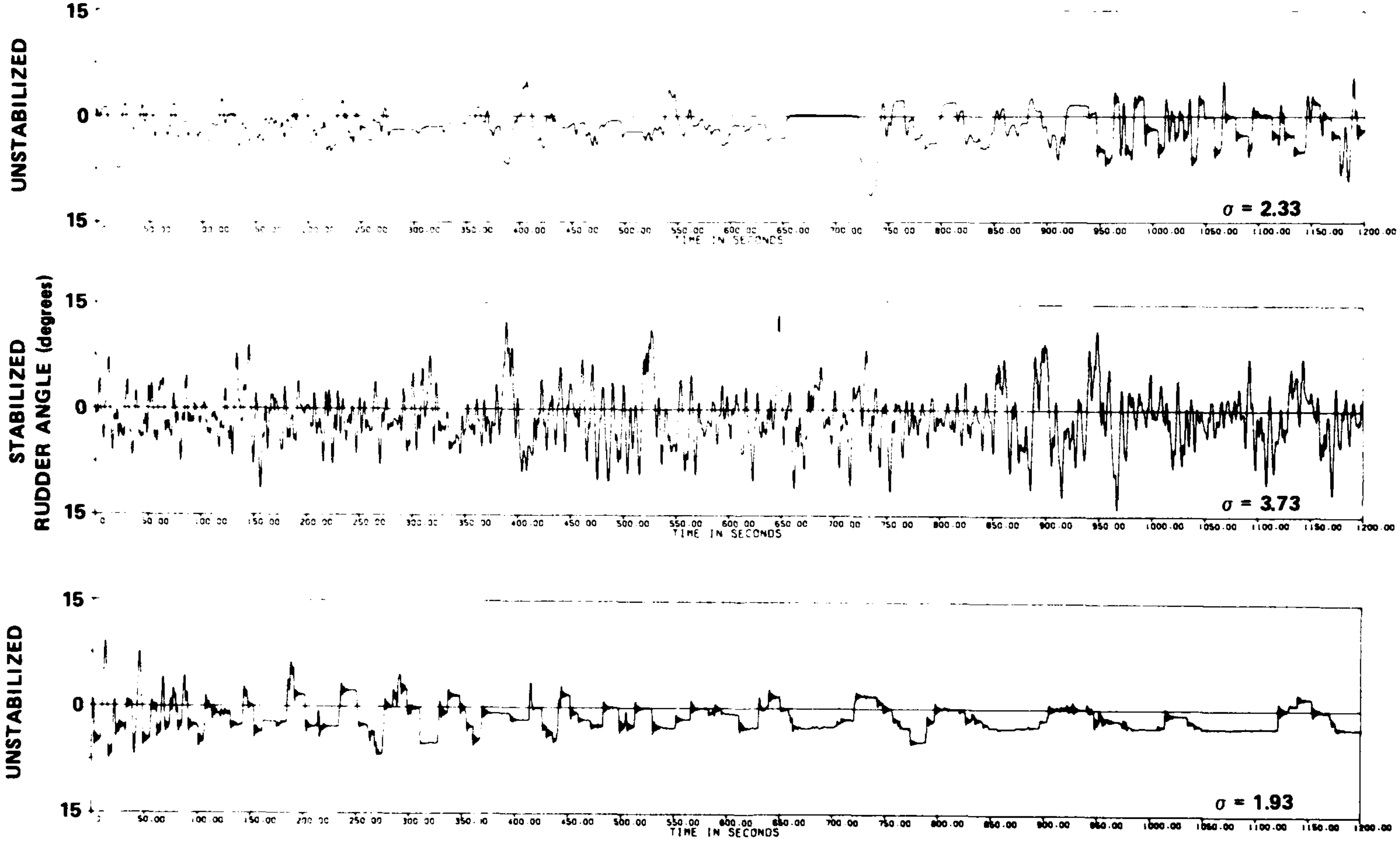


Figure A6 - Measured Rudder Angle

USCGC MELLON ROLL STABILIZATION TRIAL
BEAM SEAS, 5–7 FEET, 15 KNOTS

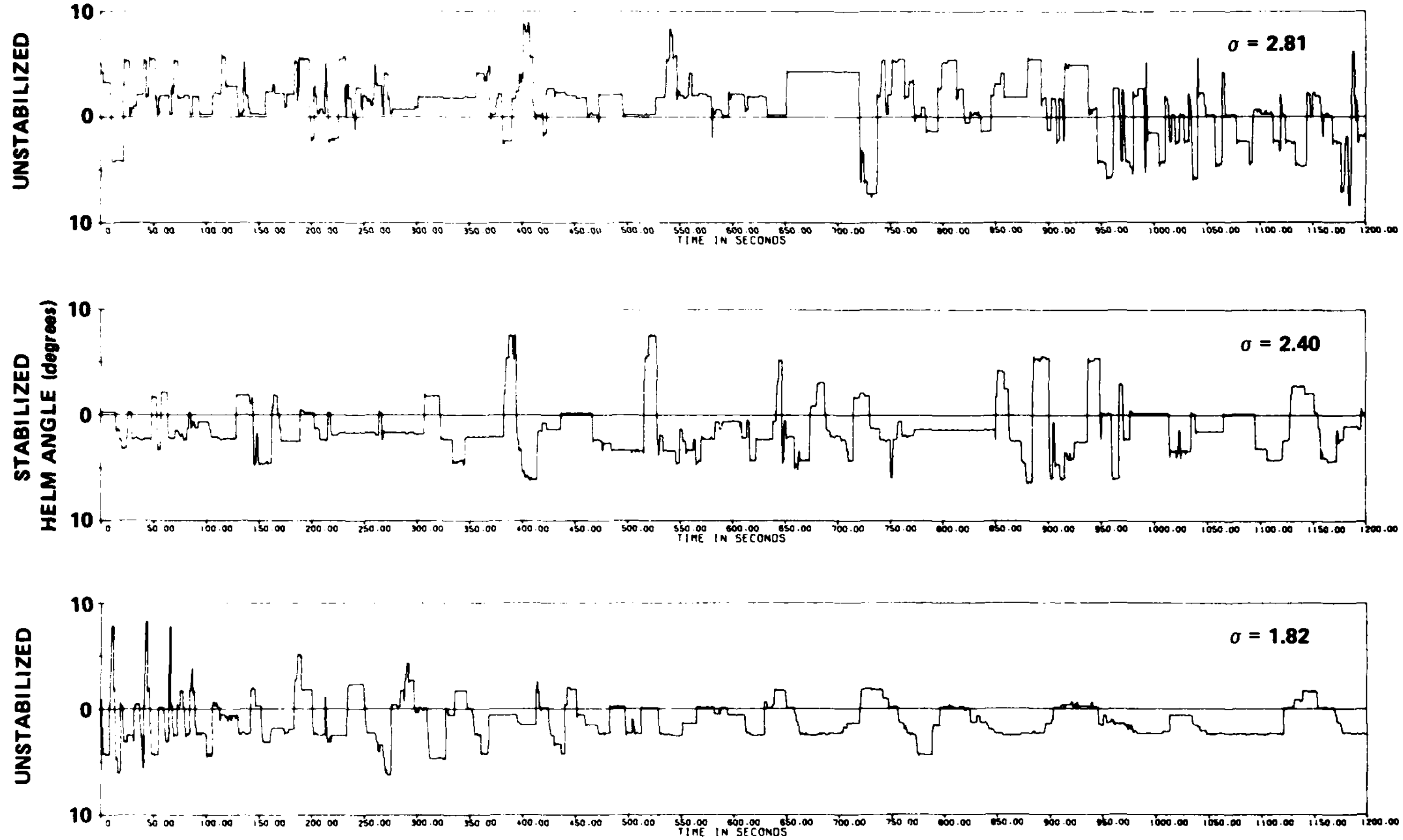


Figure A7 - Measured Helm Angle

APPENDIX B
RRS SYSTEM SIMULATION

SIMULATION PROCEDURE

Figure B1 illustrates schematically how the physical RRS components were integrated into a complete ship and rudder machinery simulation. The entire RRS hydraulic and electronic system (the bridge control unit, the electronic controller and the swash plate actuator) was tested as a total functioning system for a period exceeding 200 hours. The ship dynamics including the wave induced roll moments or wave excitations W_w that produce roll were modeled using Conolly's single degree of freedom equation. A more detailed description of this ship dynamics model, which is frequently used in preliminary or feasibility testing for antiroll tanks, has been published by Conolly.[12]

Rudder induced roll moments W_r were modeled using trial results of forced roll experiments with the rudders of USCGC HAMILTON. The hydrodynamic model of the rudder moments employed the lowest measured values of rudder lift to ensure that the predicted roll reduction performance would be conservative.

The electronic controller was mounted on the oscillating roll table and subjected to roll motions developed by the ship dynamics model. This model in turn was driven by time histories of the predetermined wave induced roll moment W_w and the roll moment W_r produced by the rudder. The roll moments produced by the rudder were developed according to the stabilizing signal δ_E. δ_E was produced by the electronic controller (experiencing the stabilized ship roll) and the swash plate actuator signal α.

The validity of the rudder dynamics model was verified by comparing simulated rudder action with measured rudder action, when measured roll and helm signals were used to drive the simulation. The results of this comparison are shown in Figure B2, where the time history of the simulated rudder action compares very well with the actual rudder action measured at sea.

SIMULATION RESULTS

All results are for irregular sea simulations only and are presented in terms of absolute RMS roll and of a percentage roll reduction without

bilge keels. For comparison, results are also provided for the ship with bilge keels. Ship heading angle relative to the waves is defined to be 180 degrees when the ship heads directly into the waves, 90 degrees when the waves approach the ship from the port beam and 0 degrees when the waves follow the ship. The range of modal wave periods and sea states shown corresponds to values which could be realistically encountered by HAMILTON Class cutters. The highest seas (Sea State 6) for which simulations were performed will be exceeded only about 4 percent of operating time on a worldwide, all-season basis.*

Sea State Effect

The effect of sea state on ship roll and available rudder moment is shown in Figure B3. This figure illustrates that the available RMS rudder induced roll moment is insensitive to sea state, whereas the wave induced roll moment increases with increasing sea severity at 16 knots in beam seas. Results were similar for other headings, and suggest that, for a given ship speed, the amount of roll reduction provided by the rudder activity is essentially constant and independent of wave height at a given ship heading.

The wave induced stabilizing moment has a much larger amplitude than the rudder induced stabilizing moment. This is true even in a Sea State 4, as shown in Figure B4. In this figure, between 1055 and 1065 seconds, the wave moment amplitude is about 0.9 million foot-pounds whereas the stabilizing moment amplitude is only about 0.2 million foot-pounds. This large difference illustrates that the rudder induced stabilizing moment must be utilized as efficiently as possible to provide significant stabilization. That is, rudder moment phasing is very important. Ideal phasing (designated as "opposed control" by Conolly[12]) would have the opposing peaks and troughs of these moments align perfectly. This has not quite been achieved with the current hardware, as is apparent in the 1055 to 1065 second range of Figure B4.

When the wave and stabilizing moment signals are examined, the wave excitations have a wide frequency range (broad banded), whereas the

---

*See Table C-1 for definition of Sea States.

stabilizing moment and the basic roll response of the ship is narrow banded. Therefore, a comparison between magnitudes of wave and stabilizing moments provides a conservative estimate of the capacity of the stabilizing moment to reduce roll.

The effect of increasing sea severity and modal wave period on RRS performance aboard a HAMILTON Class cutter is illustrated in Figure B5. The RRS system percentage roll reduction tends to decrease with increasing sea state, as would be expected due to the insensitivity of available rudder moment to sea state. For example, the 30 percent roll reductions in Sea State 4 decrease to between 12 and 25 percent in Sea State 6. The percentage roll reductions tend to increase with increasing modal wave period (increasing wavelength), although the absolute RMS roll reductions tend to decrease. The absolute RMS reductions tend to be maximum at the shorter and more frequent 7-second modal period seas and decrease with increasing period or wavelength. The improvement in the absolute RMS roll reduction at the shorter periods appears to be noticeable at Sea State 5.

The results from Sea States 4 and 5 predict that the RRS system will attain a 30 percent roll reduction, which is comparable to that attained by bilge keels under the same conditions. Conservative RRS system performance predictions indicate that the RRS system installed on a ship with bilge keels should produce roll reductions on the order of 18 to 20 percent in Sea State 4 and 5. Unlike the bilge keels, the RRS system becomes somewhat more effective at reducing roll at longer periods.

Heading Effect

The effect of heading on RRS system performance at a ship speed of 16 knots is illustrated for Sea States 4 and 5 in Figure B6. The absolute RMS results show that, for shorter period seas, the roll stabilization attained by the RRS system decreases substantially as the heading varies from quartering seas to bow seas. A maximum of absolute roll reduction occurs aft of the beam. As modal wave periods become longer, the pronounced difference between bow and quartering sea roll reduction decreases.

Speed Effect

The effect of speed on RRS system performance is demonstrated in Figure B7 for the frequently occurring Sea State 4. Simulations for speeds ranging from 12 to 20 knots at various headings and modal wave periods are presented. Speeds up to 16 knots are typical cruising or operational speeds, whereas the 20-knot speed represents the infrequent Search and Rescue or dash speed mission condition. Results indicate that maximum stabilization is attained at 16 knots with both the lower and higher speeds resulting in noticeably less roll stabilization. This speed-dependent behavior of the RRS system has been verified by MELLON during operations in the Summer of 1978 and by JARVIS during the Fall 1979 Alaskan patrol.

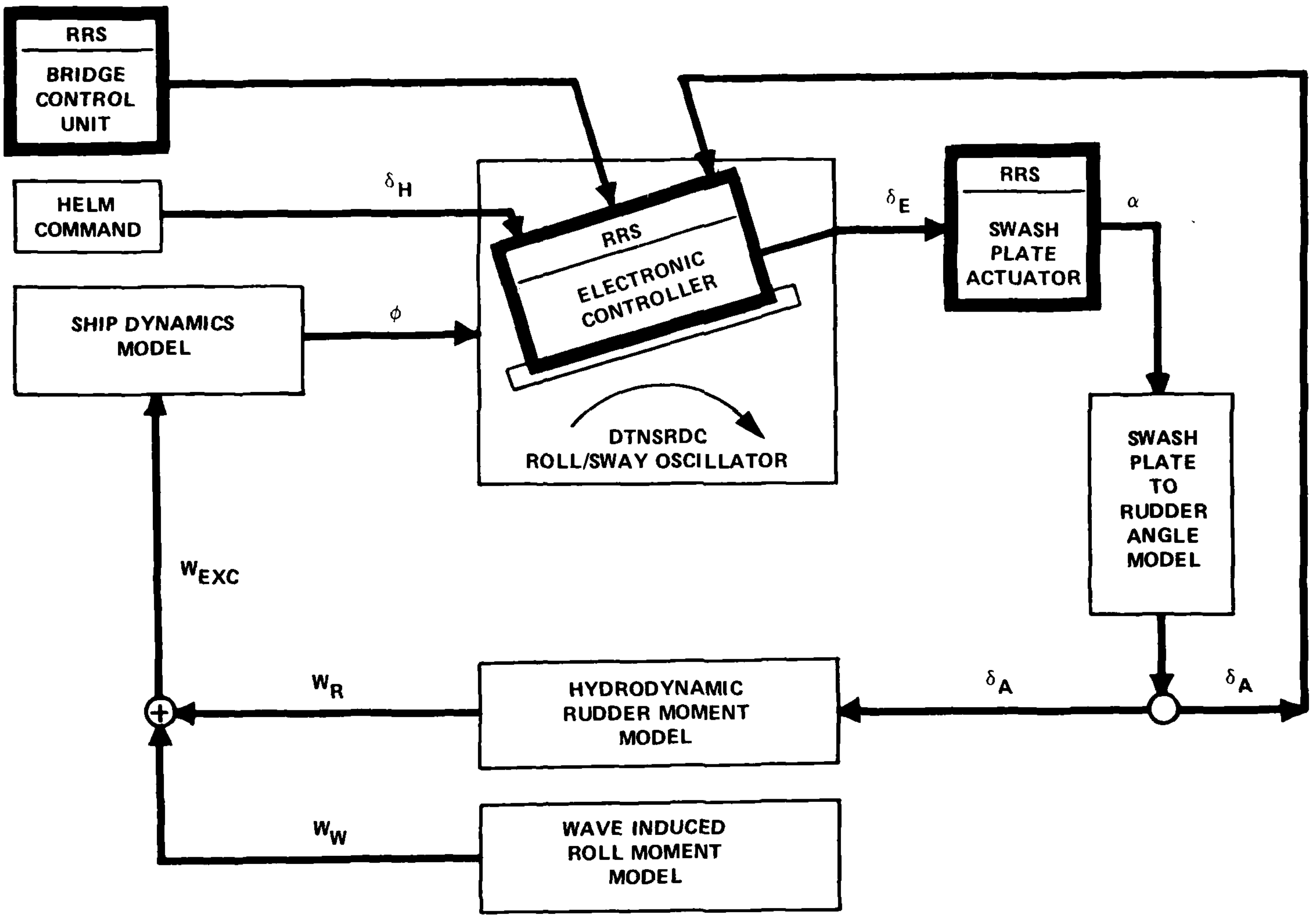


Figure B1 - RRS Performance Simulation and Component Reliability Test Set Up

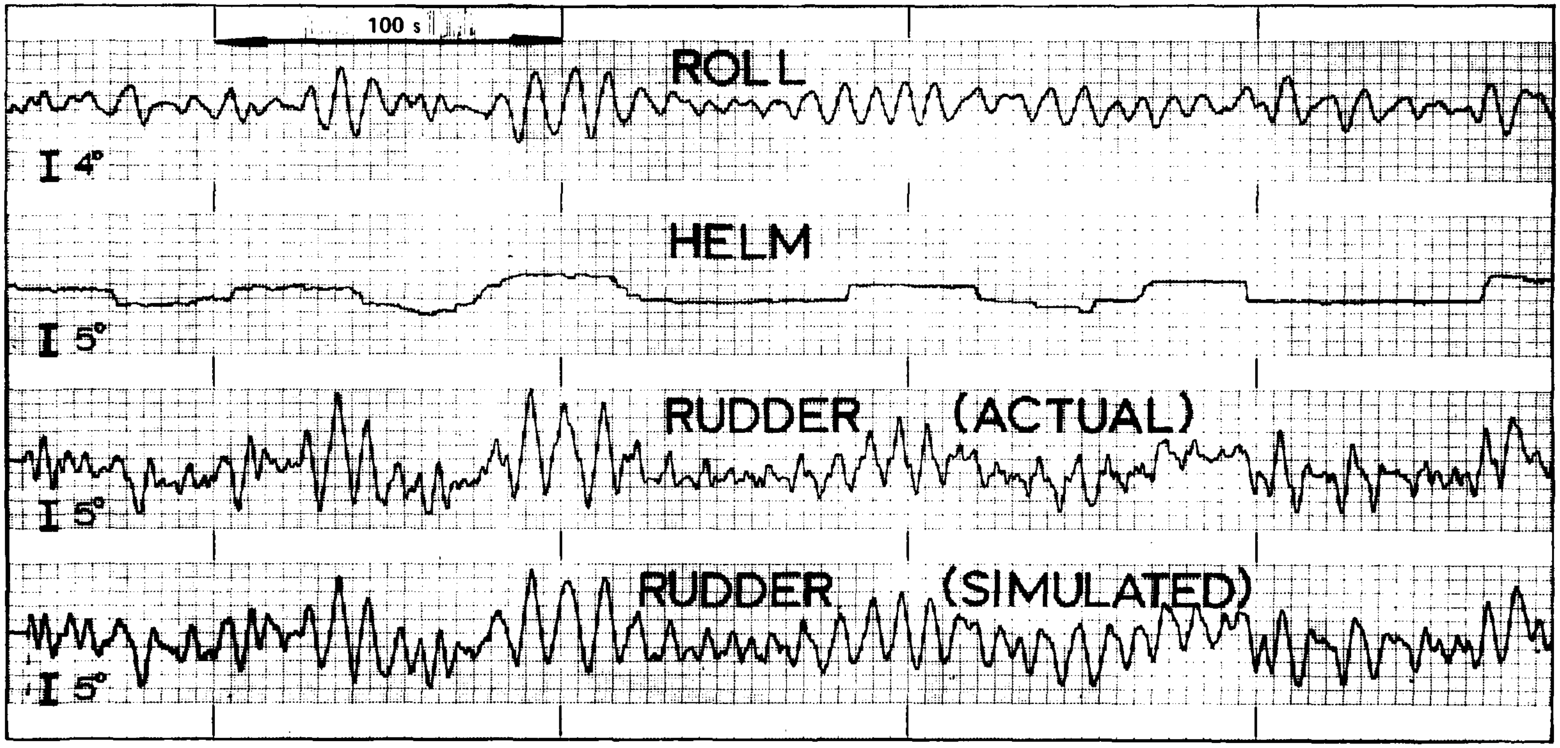


Figure B2 - Comparison between Measured and Simulated Rudder Activity

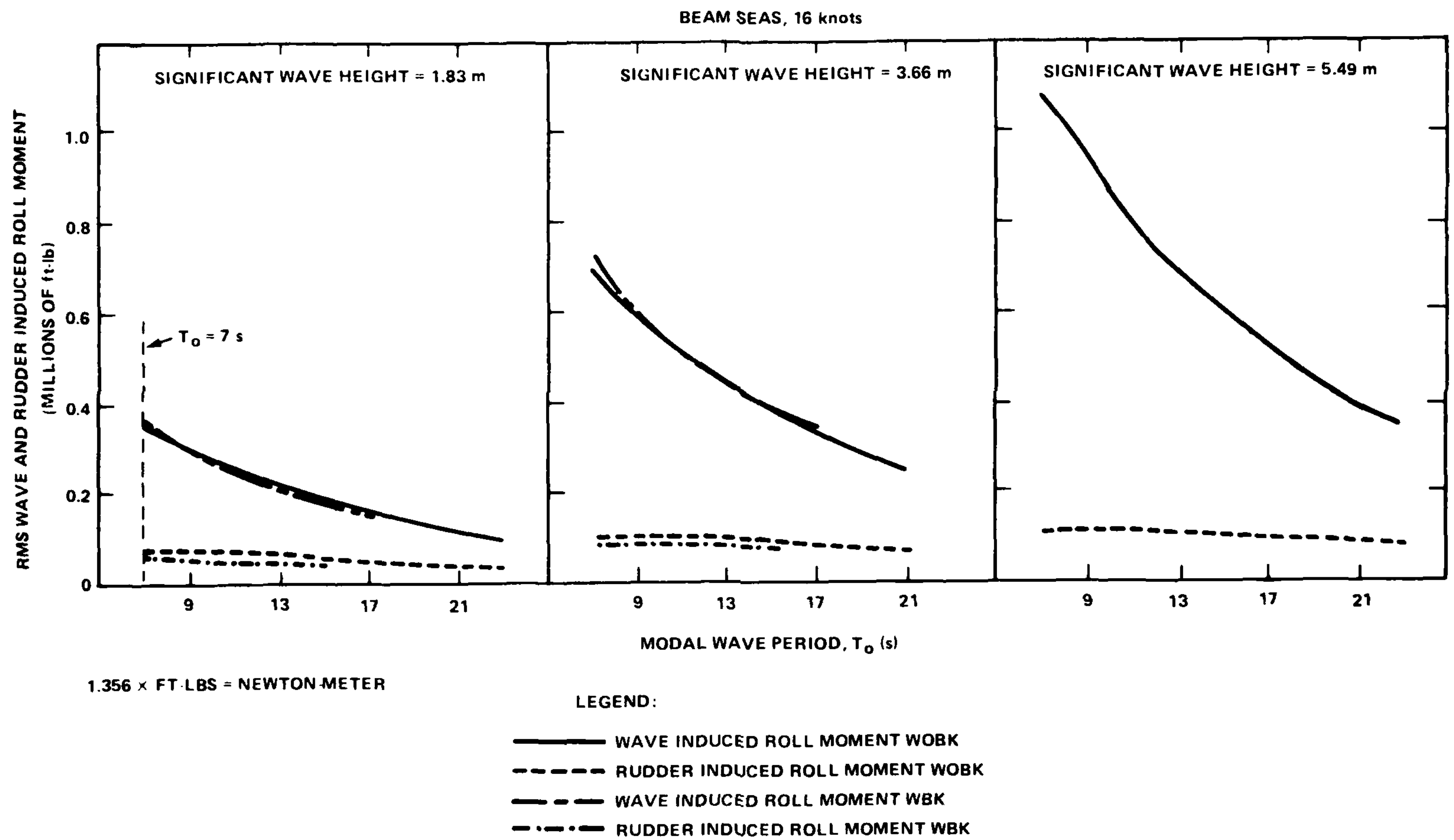


Figure B3 - Effect of Sea State on Stabilizing Roll Moment Developed by Rudder

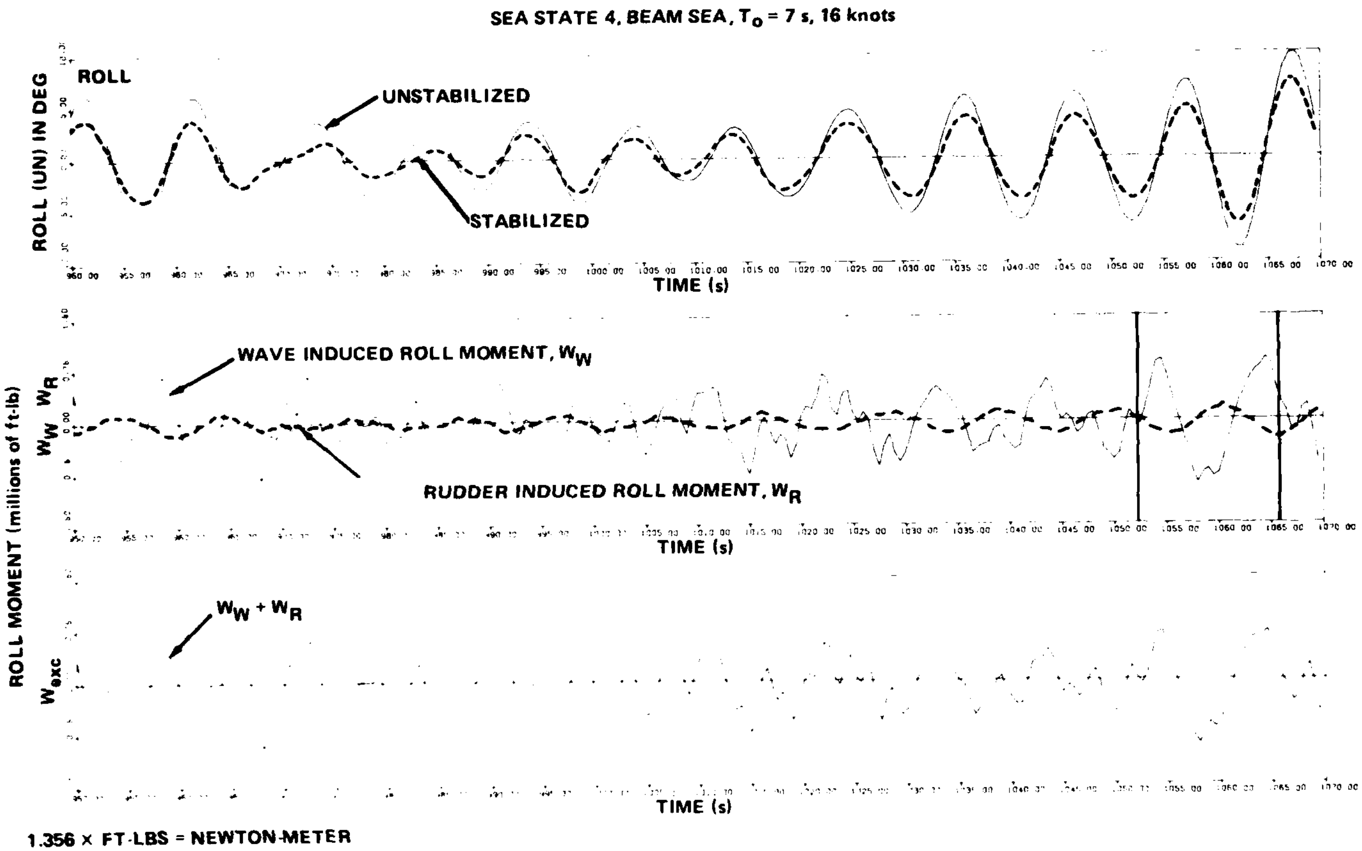


Figure B4 - Comparison of Stabilized and Unstabilized Roll and Corresponding Excitation Moments

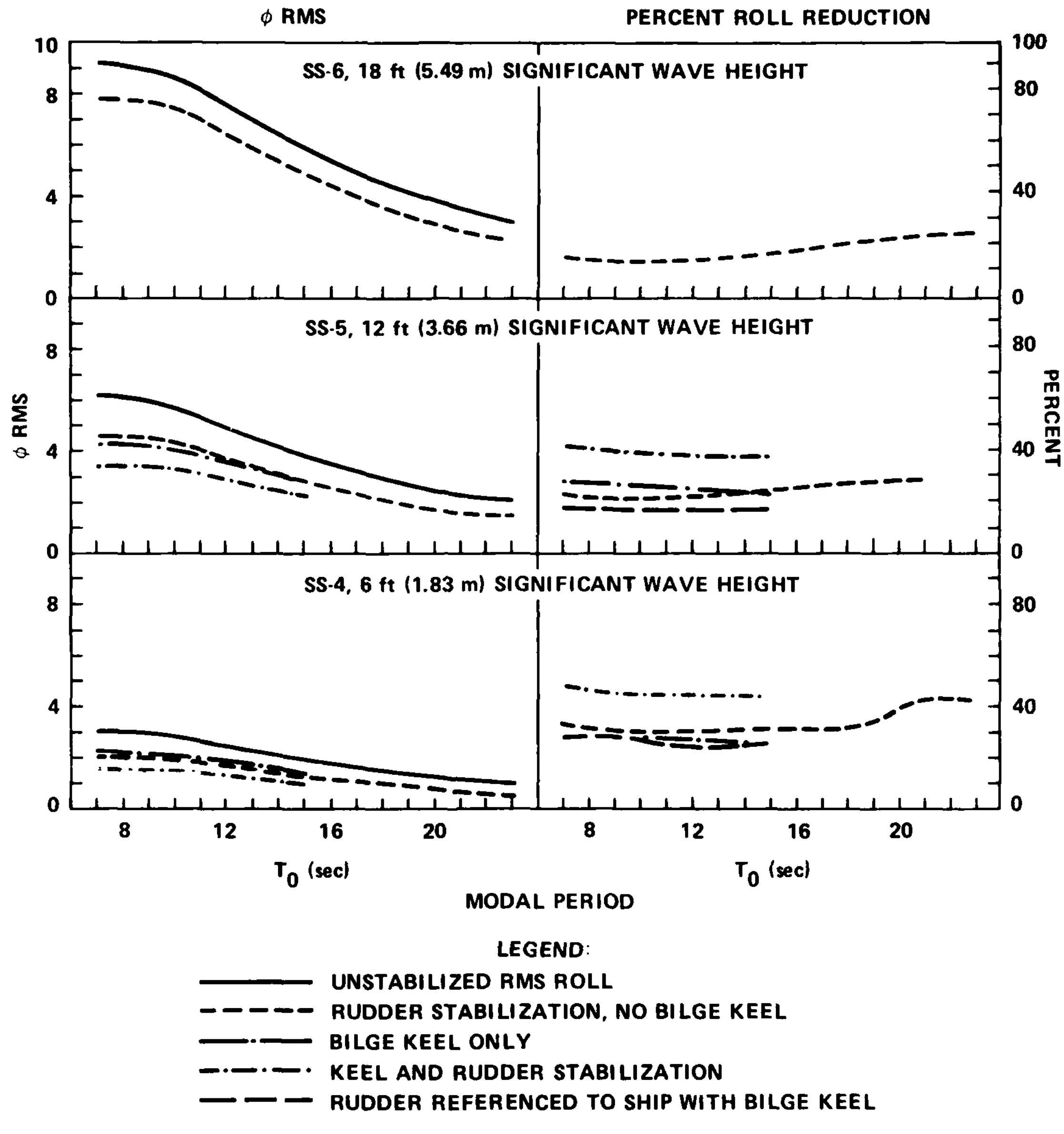


Figure B5 - Impact of Increasing Sea State and Modal Period on Predicted RRS Performance, Beam Seas, 16 Knots

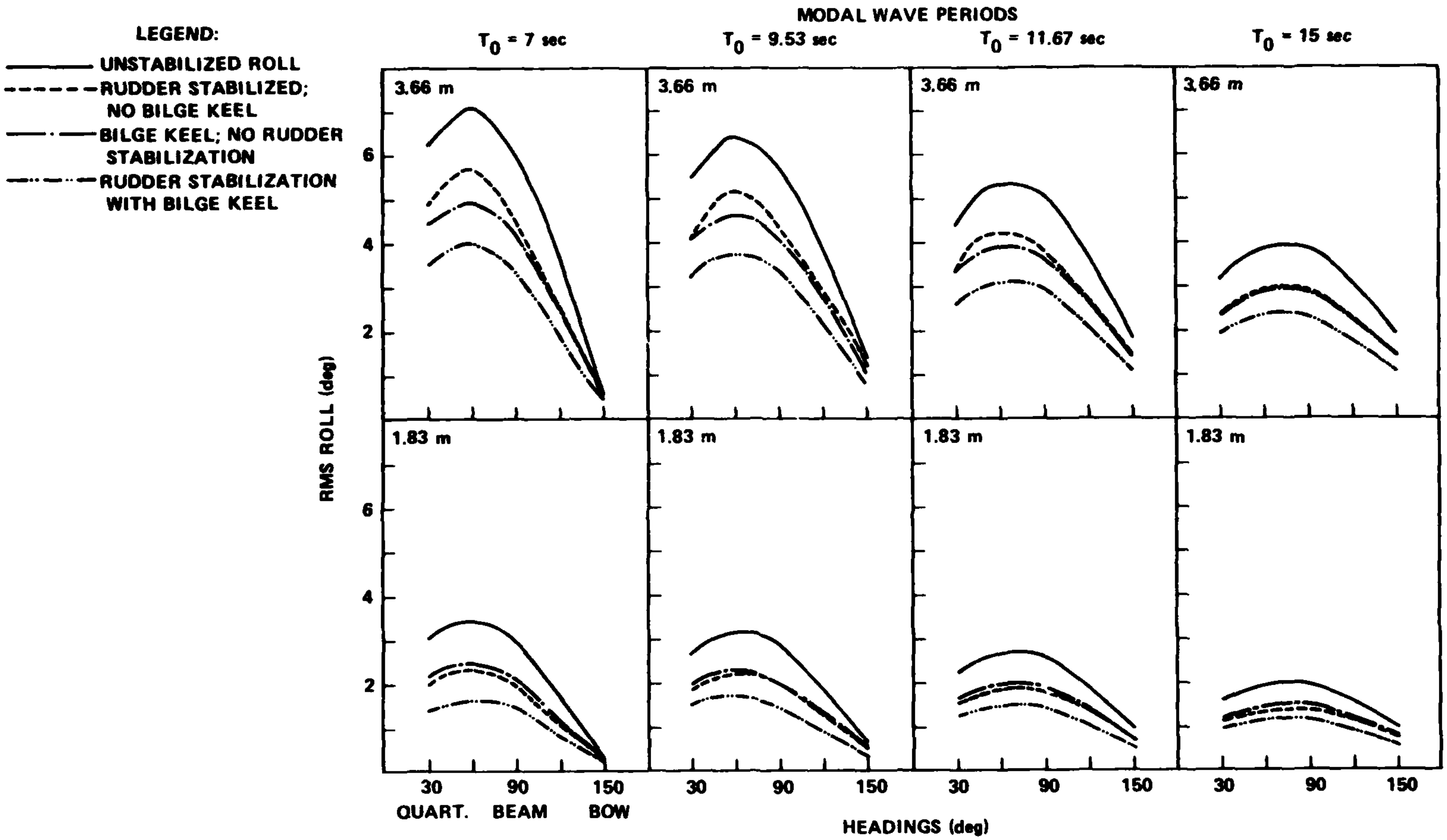


Figure B6 - Heading Effect on RRS Performance at 16 Knots

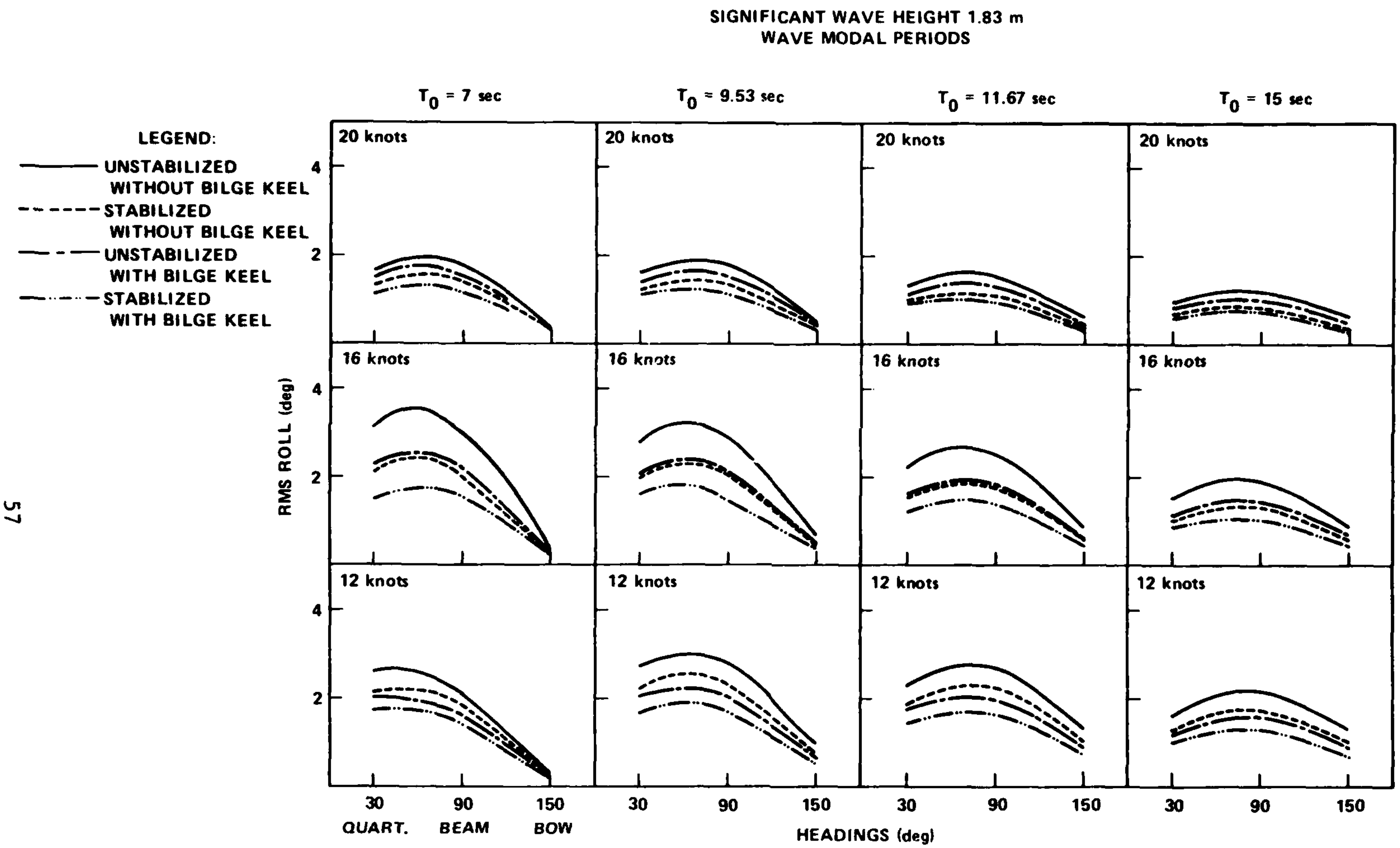


Figure B7 - Speed Effect on RRS Performance in Sea State 4

APPENDIX C
SEA REPRESENTATION FOR THE RRS SYSTEM SIMULATION

Once an RRS system is designed, built and installed aboard a ship, only three basic parameters affect the operational performance of the system. These parameters are the ship speed, heading and the seas encountered. In order to examine the general performance of RRS systems over a large variation of these parameters, the sea state simulation must be selected with some care. For the RRS performance simulations, the sea state model that was chosen had been developed over the last few years at DTNSRDC.[13-20] This model, developed as part of both USCG and subsequent U.S. Navy[14,18] projects, has been shown to be applicable for both mild and heavy seas.

Actual seas are composed of a mixture of locally generated wind waves and one or more sets of swells. Sea and swell differ substantially both in frequency and directionality characteristics. Seas developed by local winds are short, choppy and possess a relatively wide range of frequencies and directions. Swells, however, are decaying waves that tend to be narrow in both frequency range and direction, due to their generation many hours in the past and a great distance away in storms. As a result, the swells are uncorrelated to the local wind direction, and hence the local short, choppy waves.

Although it is conceptually possible to model various combinations of sea and swell,[16] this is not necessary, because a technique is available which permits the use of a sea model that accurately defines the range of ship responses likely to be produced by the almost limitless sea conditions that a ship may encounter.

The sea model consists of the two parameter wave spectra due to Bretschneider,[21]

$$S_\zeta(\omega) = \left[483.5/(\omega^5 T_o^{\,4})\right] \cdot (\tilde{\zeta}_w)_{1/3}^{\,2} \cdot \exp\left(-1944.5/(T_o^{\,4}\omega^4)\right)$$

where $S_\zeta(\omega)$ = spectral density function

$(\tilde{\zeta}_w)_{1/3}$ = significant wave height in feet

ω = wave frequency in radians per second

T_o = modal wave period in seconds

which is applicable to fully as well as partially developed wind generated seas, combined with directional wave energy spreading according to the cosine squared spreading function b_j defined by,[22,23]

$$b_j = \left[\frac{2}{\pi} \int_{\chi_j - \frac{\Delta\chi}{2}}^{\chi_j + \frac{\Delta\chi}{2}} \cos^2 \chi_j d\chi\right]^{1/2}$$

Here, b_j is an amplitude factor to account for the angular spread of wave energy, and χ_j is the wave direction referenced to the predominant wave direction.

The two parameters defining the wave spectrum are the significant wave height, $(\tilde{\zeta}_w)_{1/3}$, and the modal wave period, T_o. Significant wave height is the average of the one-third highest waves, and corresponds approximately to the value that an observer at sea would designate as the observed wave height. The modal period refers to the peak of the wave spectrum in the frequency domain, that is, the dominant waves in the seaway. Figure C1 shows the conditional probability of occurrence of modal wave periods for four different wave height ranges, while Table C1 defines sea states in terms of a range of significant wave height and a range of the shortest associated modal wave period.

The effect of variations in the frequency content of the seas (that is, the shapes of the wave spectra) was determined for the simulations by calculating the wave induced roll excitations at three different significant wave heights (6, 12, 18 feet) for a series of Bretschneider wave spectra with modal wave periods ranging from 7 to 23 seconds. Simulation results were presented for only the more common short crested seas.

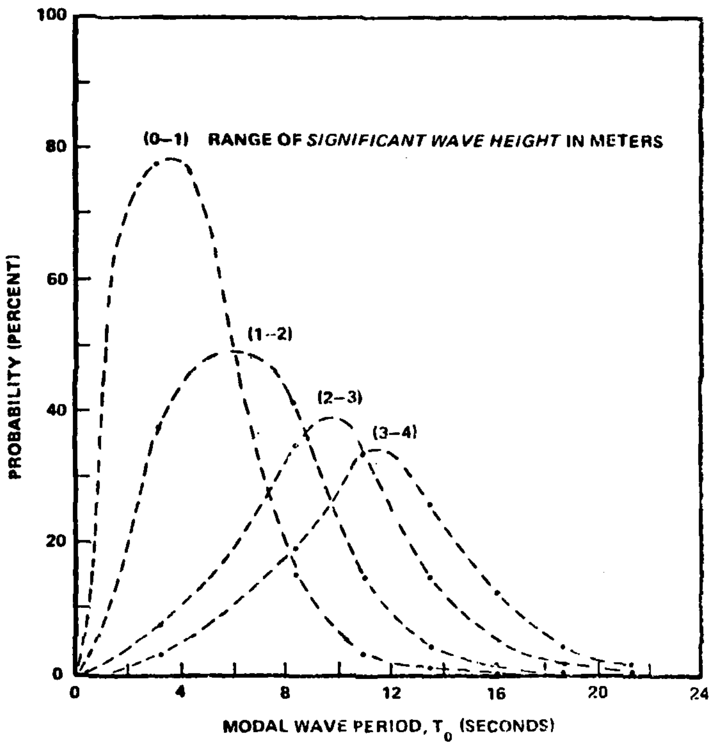


Figure C1 - World Wide All Seasons Modal Wave Period Distributions

TABLE C1

Definition of Sea States*

STATE	RANGES OF SIGNIFICANT WAVE HEIGHTS ($\tilde{\zeta}_{1/3}$) FT	RANGES OF MODAL WAVE PERIODS T_0 SEC
1	0 – 1.92	0 – 3.08
2	1.92 – 4.13	3.08 – 4.52
3	4.13 – 5.66	4.52 – 5.29
4	5.66 – 7.35	5.29 – 6.03
5	7.35 – 13.04	6.03 – 8.03
6	13.04 – 20.80	8.03 – 10.15
7	20.80 – 40.33	10.15 – 14.13
8	40.33 – 61.58	14.13 – 17.45

NOTE: 1. T_0 *periods corresponding to the steepest, partially developed wind-generated* waves, short fetch, high wind, moving hurricane.

2. Steeper waves do occur, but thet are rare and are generally associated with land locked bays or lakes.

3. $T_0 = [(\tilde{\zeta}_{1/3})/0.202]^{1/2}$ — Modal period of partially developed hurricane sea (Bretschneider).

4. $T_0 = [(\tilde{\zeta}_{1/3})/0.127]^{1/2}$ — Modal period of fully developed wind sea (Pierson-Neumann-James).

5. $\lambda_0 / (\tilde{\zeta}_{1/3}) = 40$ — Pierson-Moskowitz wave spectra

6. $\lambda_0 / (\tilde{\zeta}_{1/3}) = 25$ — Bretschneider

7. $\lambda_0 / (\tilde{\zeta}_{1/3}) = 10$ — Steepest observed, Hogben and Lumb

λ_0 = Wavelength corresponding to period of spectrum peak, T_0

* From Reference 22

APPENDIX D
ESTIMATES OF THE EFFECTIVENESS OF STEERING MOTOR COOLING
by
T. McNamara

Measurements of starboard steering motor case temperature were made aboard JARVIS after a shroud and three 65 cfm fans were installed for added cooling. These measurements were used to estimate the total heat flux from the motor while the RRS system was operating. Once the total heat flux had been determined, calculations were performed to estimate the change in case temperature for different cooling configurations. The assumptions involved in these calculations were as follows:

Emissivity:	0.78 (gray paint)
	0.98 (black paint)
Total heat flux:	2400 watts (constant for all cases)
Ambient temperature:	85°F (North Pacific)
	110°F (Tropical)
Geometry:	Simple cylindrical shape, no end effects

Five different cooling conditions were analyzed:

1. Original (1/8" thick gray paint) - HAMILTON Class cutters currently use two 30 hp 30-minute duty cycle steering motors, each covered with a thick (approximately 1/8 inch) coating of gray paint. The only cooling comes from a small internal fan which provides very little circulation.

2. Shroud, Small Fan (three 65 cfm fans, gray paint) - To increase cooling aboard JARVIS, shrouds were installed over the back half of each motor in September 1979, and three 65 cfm cooling fans were mounted at the back end of each shroud.

3. Shroud, Large Fan, Flat Black Paint (old paint removed) - JARVIS temperature measurements in September 1979 indicated little change due to the shroud and fans, so larger, 200 cfm fans were installed in shrouds aboard MELLON in October 1979. Also, the old paint was chipped away, and a thin coating of thermally conducting flat black paint was applied.

4. Shroud, Large Fans (three 200 cfm fans, gray paint) - Calculations were made to estimate the effect of the shroud and large fans with the original paint.

5. <u>Flat Black Paint Only</u> (old paint removed, no shroud) - Additional calculations were made to estimate the effect of the flat black paint only, with no shroud or fans.

For each condition, estimates were made of the casing surface temperature in tropical waters and in the North Pacific.

The results for North Pacific waters are shown in Figure D1. The 30 horsepower steering motors are rated for a 90°F temperature rise above ambient, as noted in the figure. The results reveal that only Cases 3 and 4 (shroud, large fans, with or without black paint) provide adequate cooling. Figure D2 shows similar results for motor operation in tropical waters. Here, Cases 3, 4 and 5 provide adequate cooling.

The results indicate that the shroud and large fans are necessary for adequate cooling, and that, in addition, the black paint is desirable to decrease the surface temperature even further. The black paint not only provides a 15°F cooler surface temperature, but it also maintains a lower internal motor temperature due to its increase in thermal conductance over a 1/8" thick coating of paint.

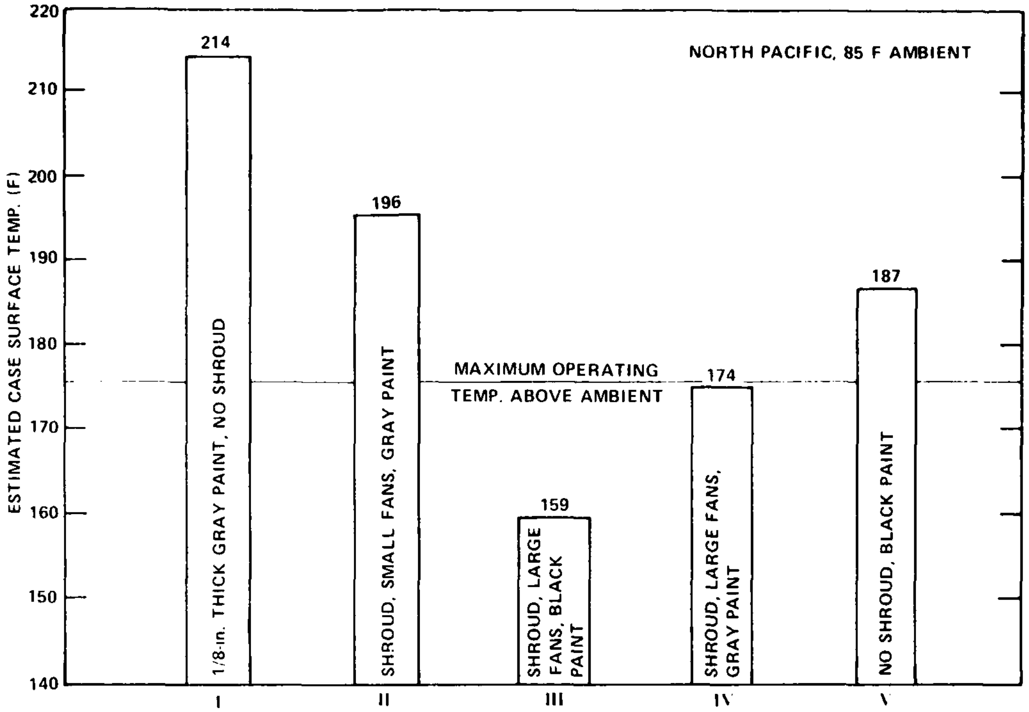


Figure D1 - Cooling Conditions, North Pacific

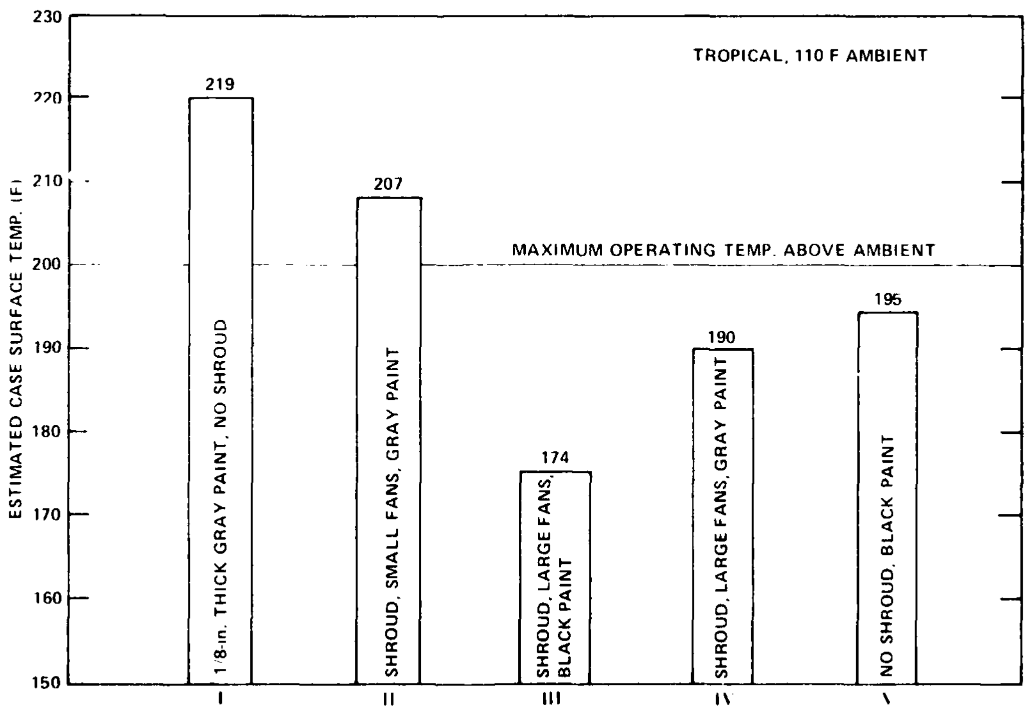


Figure D2 - Cooling Conditions, Tropical

REFERENCES

1. Cowley, W.E. and T.H. Lambert, "The Use of the Rudder as a Roll Stabilizer," Proceedings Third Ship Control Systems Symposium (1972).

2. Cowley, W.E. and T.H. Lambert, "Sea Trials on a Roll Stabilizer Using the Ship's Rudder," Proceedings Fourth Ship Control Systems Symposium (1975).

3. Lloyd, A.R.J.M., "Roll Stabilization by Rudder," Proceedings Fourth Ship Control Systems Symposium (1975).

4. Carley, J.B., "Feasibility Study of Steering and Stabilizing by Rudder," Proceedings Fourth Ship Control Systems Symposium (1975).

5. Pinegar, F.A. and H.G. Kolwey, "HH-3F Helicopter/USCGC HAMILTON (WHEC-715) Dynamic Interface Evaluation," Naval Air Test Center Report RW-2GR-76 (May 1976).

6. Baitis, A.E. and W.G. Meyers, "Progress Report on the NSRDC Anti-Roll Tank Facility," Seventeenth American Towing Tank Conference (1974).

7. Whyte, P.H., "A Note on the Application of Modern Control Theory to Ship Roll Stabilization," Proceedings of the Eighteenth General Meeting of the American Towing Tank Conference (1977).

8. Baitis, A.E., G.G. Cox and D.A. Woolaver, "The Evaluation of VOSPER Active Fin Roll Stabilizers," Proceedings Third Ship Control Systems Symposium (1972).

9. Cox, G.G. and A.R. Lloyd, "Hydrodynamic Design Basis for Navy Ship Roll Motion Stabilization," Transactions Society of Naval Architects and Marine Engineers, Vol. 85 (1977).

10. Zarnick, E.E. and J.A. Diskin, "Modelling Techniques for the Evaluation of Anti-Roll Tank Devices," Proceedings Third Ship Control Systems Symposium (1972).

11. Naval Sea Systems Command, Contract Specifications, "Fin Stabilization System (FFG-7)," SHIPS-S-5832 (5 Aug 1977).

12. Conolly, J.E., "Rolling and Its Stabilization by Active Fins," Transactions of the Royal Institute of Naval Architects, Vol. III (1969).

13. Baitis, A.E. et al., "Design Acceleration and Ship Motions for LNG Cargo Tanks," Tenth Symposium on Naval Hydrodynamics (Jun 1974).

14. Baitis, A.E. et al., "LNG Cargo Tanks: A Ship Motion Analysis of Internal Dynamic Loadings," GASTECH 74, International LNG and LPG Congress, Amsterdam (Nov 1974).

15. Bales, S.L. et al., "Rigid Body Ship Responses and Associated Periods for a Series of LNG Ships," Naval Ship Research and Development Center Report SPD-517-04 (Apr 1975).

16. Baitis, A.E. et al., "Summary of Development for LNG Tank Design Acceleration Rules," Naval Ship Research and Development Center Report SPD-517-03 (Dec 1976).

17. Baitis, A.E. et al., "Preliminary Roll and Pitch Predictions for Two Candidate Hull Forms of T-AGS," Naval Ship Research and Development Center Report SPD-576-01 (Aug 1974).

18. Bales, S.L., "Ship Motion Predictions for the MONOB1, Operating in Waters Near the Bahamas," Report DTNSRDC/SPD-727-01 (Sep 1976).

19. Baitis, A.E. et al., "A Non-Aviation Ship Motion Data Base for the DD-963, CG-26, FF-1052, FFG-7 and the FF-1040 Class Ships," Report DTNSRDC/SPD-738-01 (Dec 1976).

20. Ochi, M.K. and S.L. Bales, "Effect of Various Spectral Formations in Predicting Responses of Marine Vehicles and Ocean Structures," Ninth Annual Offshore Technology Conference, Report OTC 2743 (1977).

21. Bretschneider, C.L., "Wave Variability and Wave Spectra for Wind Operated Gravity Waves," Department of the Army Corps of Engineers Technical Memorandum 118 (1959).

22. Baitis, A.E. et al., "A Seakeeping Comparison Between Three Monohulls, SWATHS, and Column-Stabilized Catamaran Designed for the Same Mission," Naval Ship Research and Development Center Report SPD-622-01 (Jul 1975).

23. Brown, R.G. and F.A. Camaratta, "NAVAIRENGCEN Ship Motion Computer Program: Theory, Documentation, and Users Guide," NAEC Report MISC-903-8.

24. Woolaver, D.A. and G.R. Minard, "Rudder Roll Stabilization System User's Guide," Report DTNSRDC/SPD-0930-01 (Jan 1980).

DTNSRDC ISSUES THREE TYPES OF REPORTS

1. DTNSRDC REPORTS, A FORMAL SERIES, CONTAIN INFORMATION OF PERMANENT TECHNICAL VALUE. THEY CARRY A CONSECUTIVE NUMERICAL IDENTIFICATION REGARDLESS OF THEIR CLASSIFICATION OR THE ORIGINATING DEPARTMENT.

2. DEPARTMENTAL REPORTS, A SEMIFORMAL SERIES, CONTAIN INFORMATION OF A PRELIMINARY, TEMPORARY, OR PROPRIETARY NATURE OR OF LIMITED INTEREST OR SIGNIFICANCE. THEY CARRY A DEPARTMENTAL ALPHANUMERICAL IDENTIFICATION.

3. TECHNICAL MEMORANDA, AN INFORMAL SERIES, CONTAIN TECHNICAL DOCUMENTATION OF LIMITED USE AND INTEREST. THEY ARE PRIMARILY WORKING PAPERS INTENDED FOR INTERNAL USE. THEY CARRY AN IDENTIFYING NUMBER WHICH INDICATES THEIR TYPE AND THE NUMERICAL CODE OF THE ORIGINATING DEPARTMENT. ANY DISTRIBUTION OUTSIDE DTNSRDC MUST BE APPROVED BY THE HEAD OF THE ORIGINATING DEPARTMENT ON A CASE-BY-CASE BASIS.